POINTS

(Precision Optical INTerferometer in Space)

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JPL POINTS STUDY TEAM

FY 91 - FY 93

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POINTS: Background

High-precision, high-throughput interferometer for astrometry only

Science compatible with astrophysics (AIM) and planet detection (TOPS-1)

Moderate-class mission: < $400M, intermediate launch vehicle

Compact, lightweight, suitable for high Earth orbit

10-year mission life

Minimal number and complexity of mechanisms or deployable parts

Formal collaborative effort between SAO and JPL since FY 90:
Total funding FY 90- FY 93: ~ $2.0 M

Source	SAO ($K)		JPL ($K)
Code S	1224		302
SAO	185		
JPL (DDF, 0SS1)	-		328
	1409	+	630 = $2.039 M

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POINTS spacecraft flight-system configuration: Drivers and trade-offs

Uniform thermal environment for instrument

Use solar panel to keep instrument in permanent shadow

Ample power reserve for long cruise (17 hr) to HEO and mission life

Make solar panel deployable; consider using GaAs instead of Si

Reduce mass, number and complexity of mechanisms

Integrate instrument and spacecraft bus; gimbal the solar panel

Maximum sky accessibility

Optimize solar panel size and distance from SC

Point instrument/spacecraft with gimballed spin and tilt, and roll about Sun-1ine

Solar-pressure torques

Simple, Sun-facing configuration permits accurate modeling, minimal degradation to orbit determination (< 0.02 microarcsec stellar aberration correction over 24 hours)

May require thrusters for momentum dumping

POINTS spacecraft flight-system configuration: Science instrument

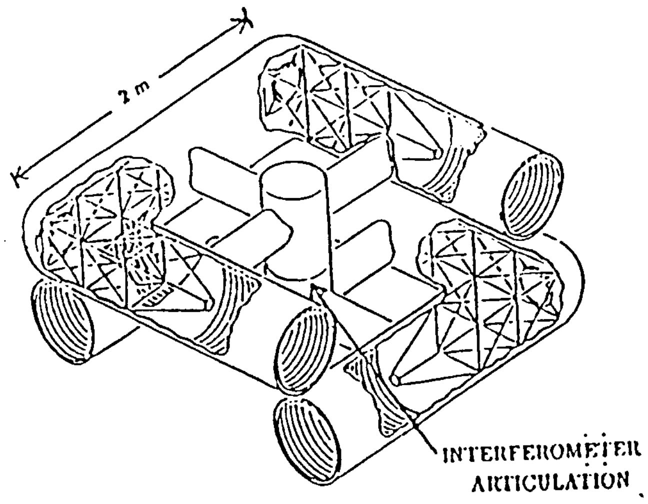
Two identical interferometers: 2-m baselines, 25-cm apertures

Optical path lengths fixed: no variable delay lines

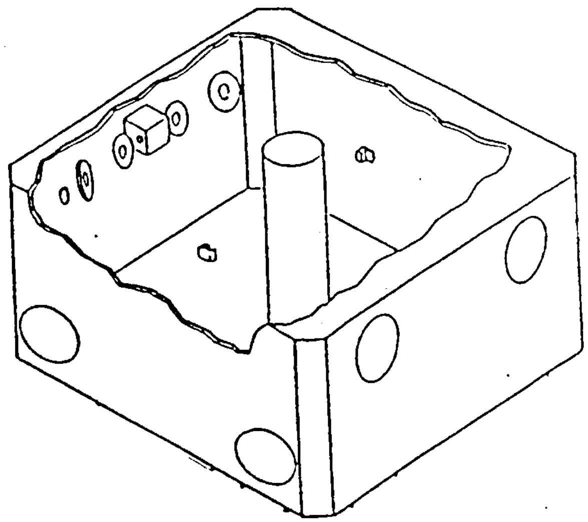
Interferometers oriented at 90 (± 3) degrees instead of collinear

- **Numerous bright reference stars (>2000 sq. deg. torus of sky)**
- **Absolute parallax without distant "stationary" references**
- **Rapid closure around sky to help estimate instrument parameters**

Original SAO design:
U-tubes and optical-bench truss

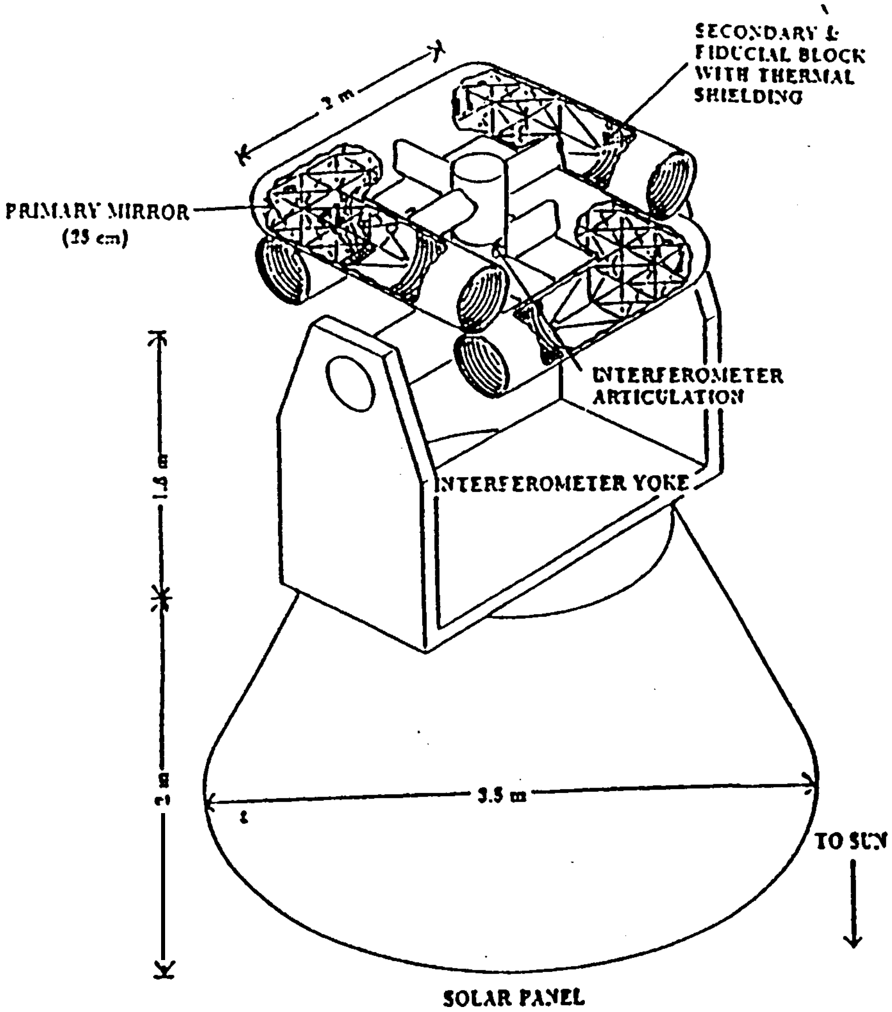


FY 92 JPL design:
No U-tubes: housing ("hatbox")
around both interferometers

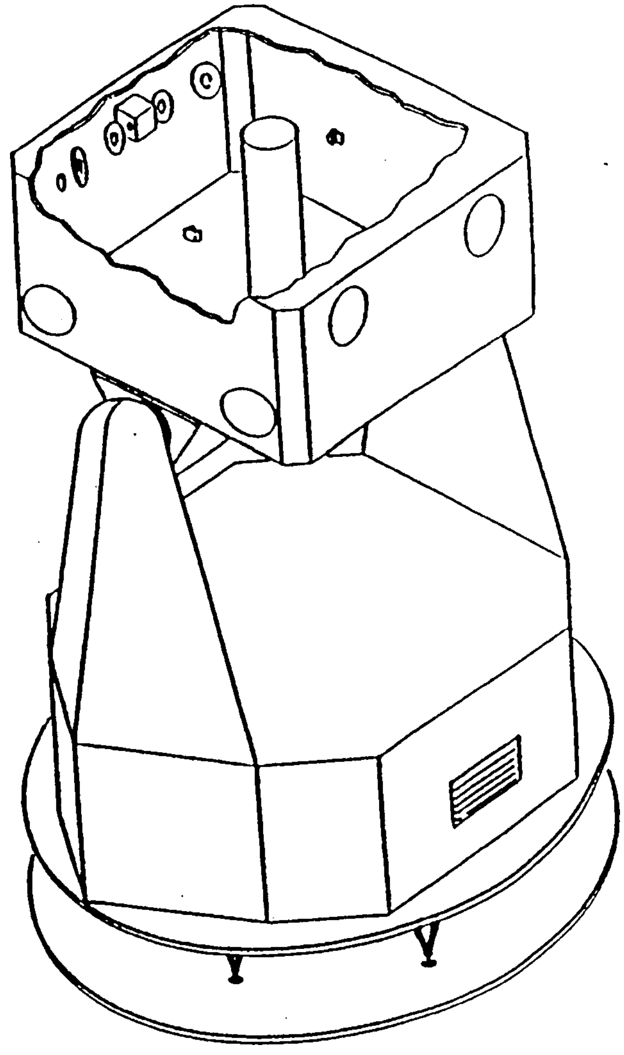


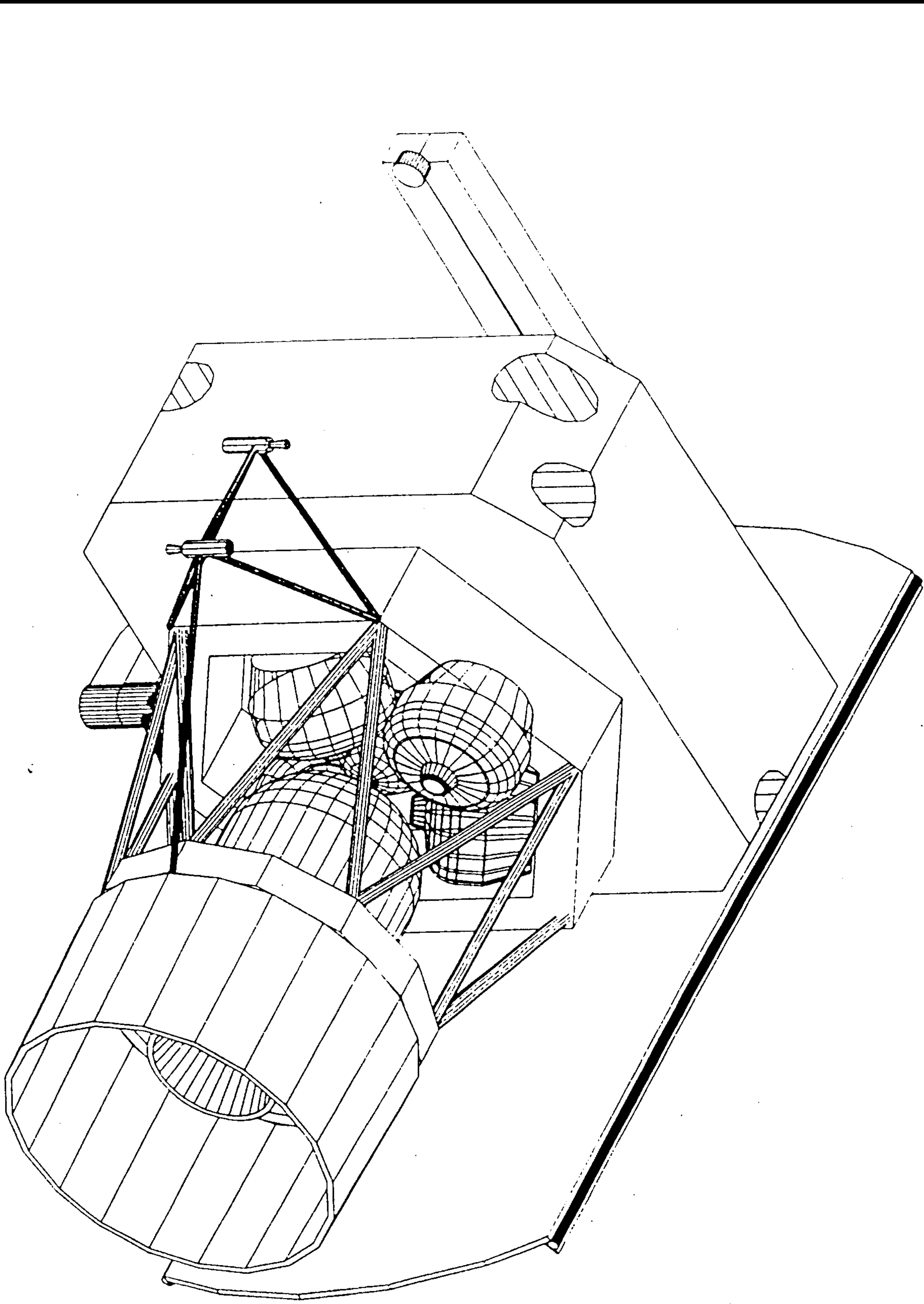
POINTS spacecraft flight-system configuration

SAO/JPL FY 91

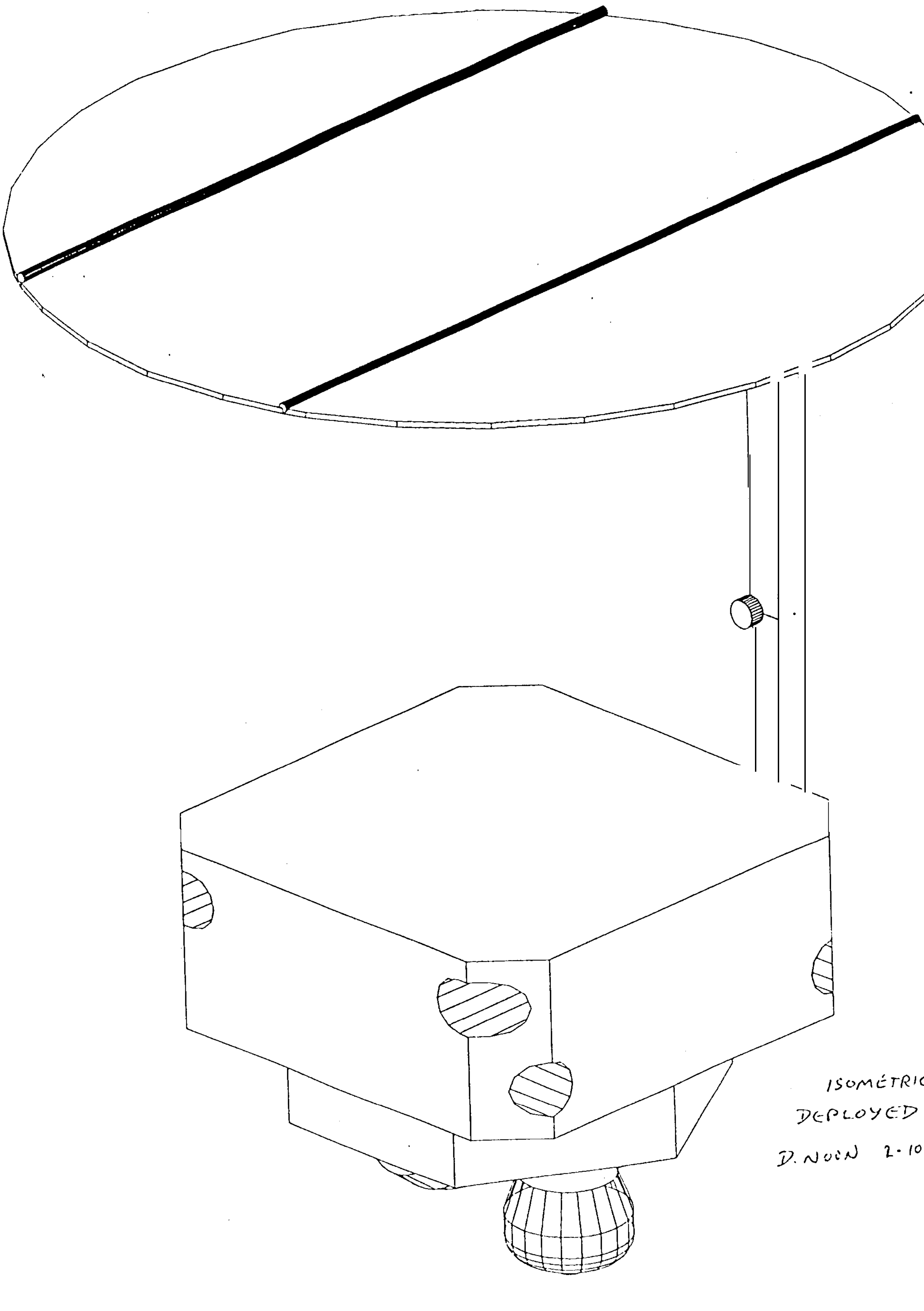


JPL FY 92





ISOMETRIC
STOWED CON FIG.
D. NOON 2-10-93



ISOMETRIC
DEPLOYED
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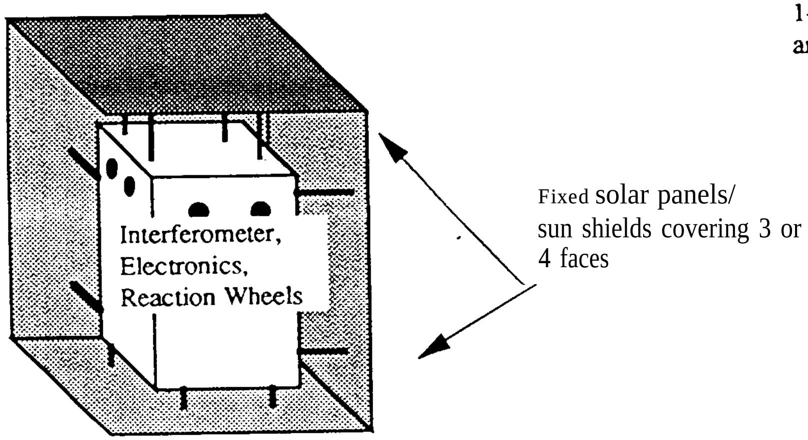
POINTS spacecraft flight-system configuration: Alternative options

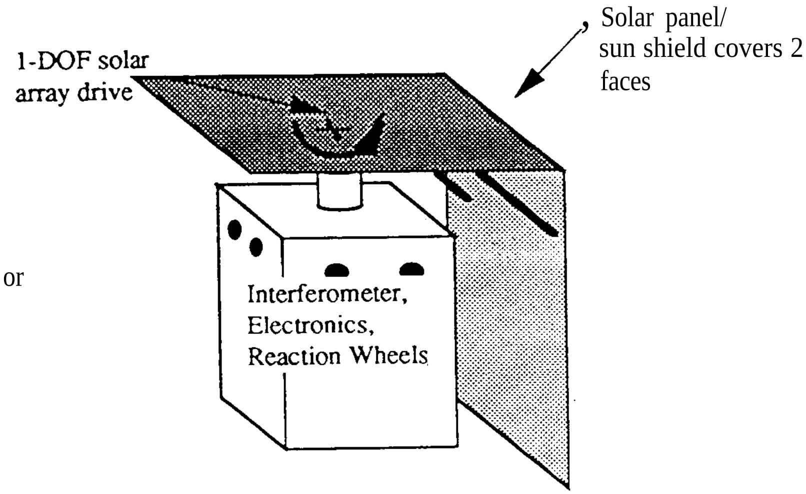
Considerations included: Sky accessibility, AACS complexity and cost, solar pressure torques, uniform thermal environment, solar power, structural dynamics, overall mass, reliability for 10-year mission life.

Option "E" (JPL FY 91, 92) has been replaced by Option " C" (JPL FY 93).

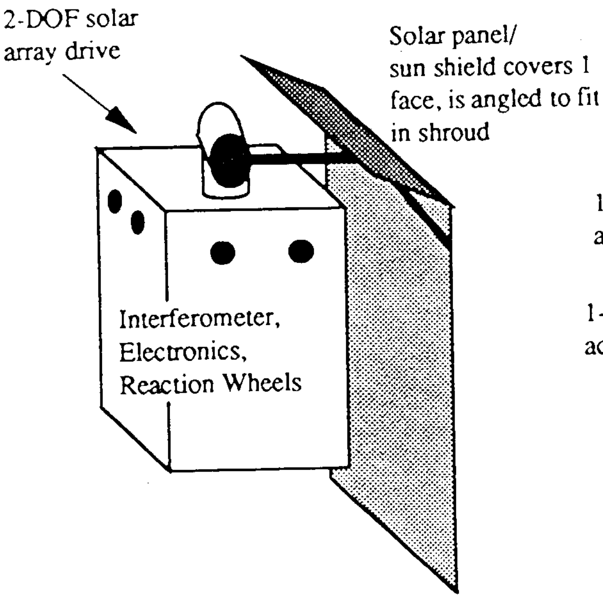
POINTS Spacecraft Configuration: Alternative Options



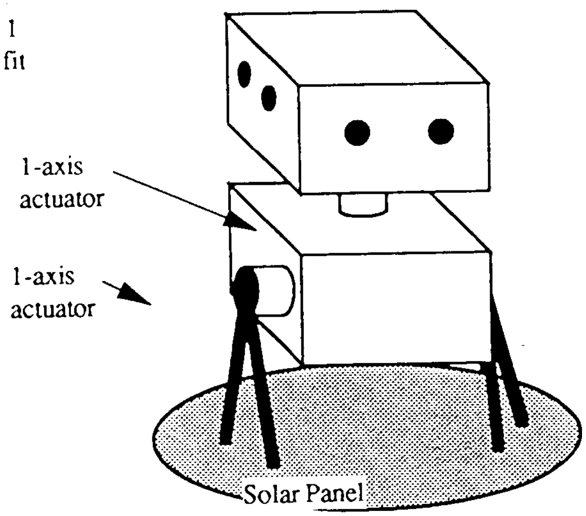
A "Borg"



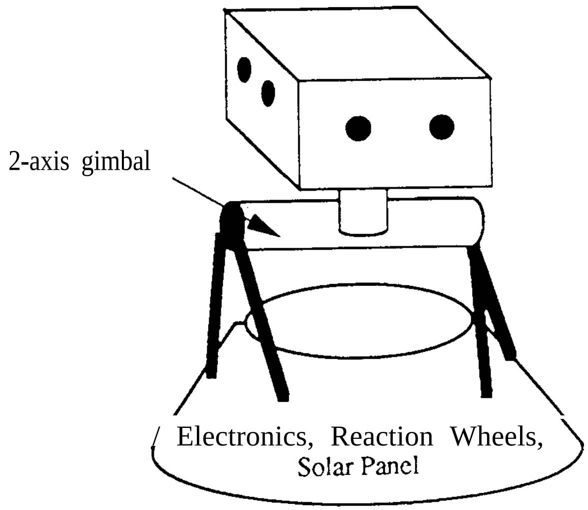
B "Semi-Borg"



C "Borg with 2 DOF Solar Panel"



D "Semi-Baseline"



E "Baseline"

Actuators for POINTS: Reaction wheeIs

Provide 3-axis stabilization

Tetrahedral arrangement (or 3 orthogonal plus 1 skewed)

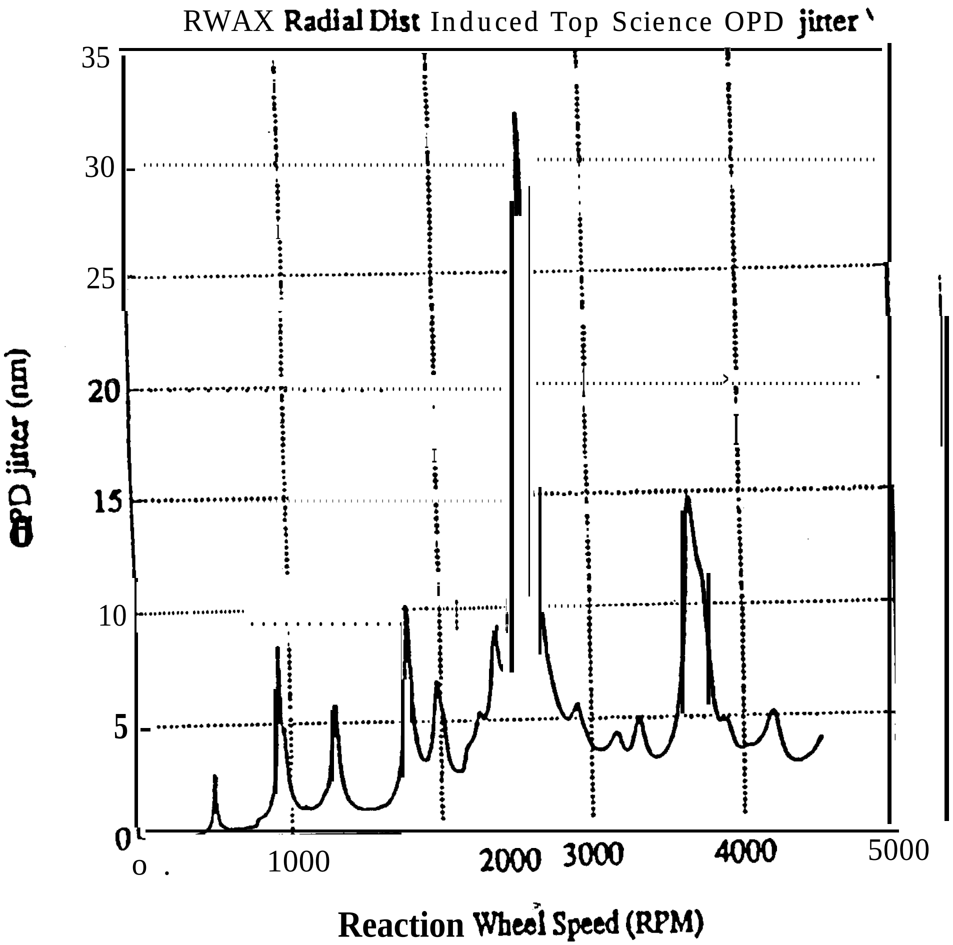
Torque capacity ~0.3 that of HST (-0.25 N-m), angular momentum capacity ~0.15 that of HST (~40 N-m-s), peak power ~150 W. (Mass not necessarily as small as ~0.15 HST wheels, because of required disturbance isolation.)

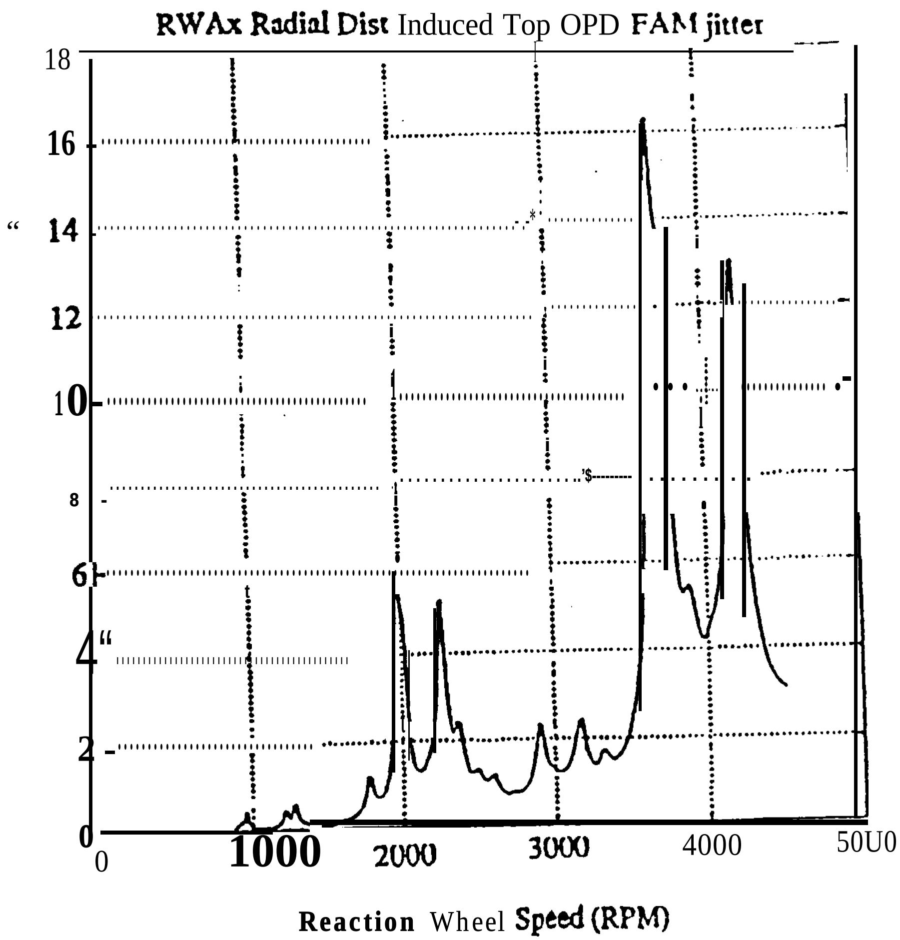
Wheels with above specs could provide slews of 10 deg in 60 see, 50 deg in 140 see, 100 deg in 210 sec.

Vibration isolation essential to avoid loss of fringe visibility. Honeywell claims to be able to provide passive damping at wheeIs better than that of HST, for wheels with above specs, and adequate to keep effect on measured interferometer OPDS under 10 pm (<1-μarcsec contribution to overall astrometric error budget).

Fluctuations in measured OPDs mused by reaction-wheels

(Based on JPL FY 92 POINTS spacecraft configuration, HST wheels with no isolation; radial disturbances shown for illustration. Work by J. Melody, JPL.)





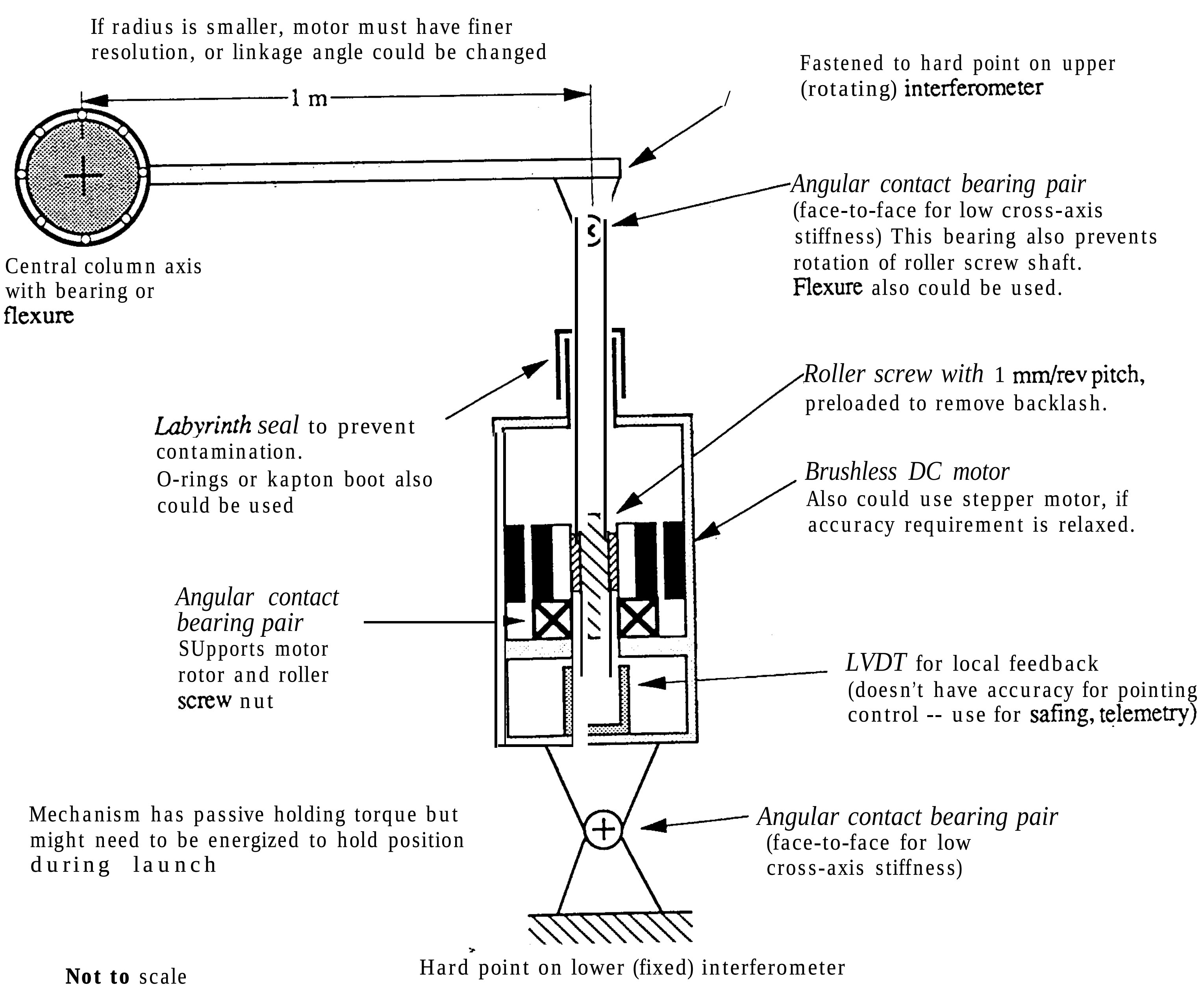
Actuators for POINTS: Interferometer relative articulation

Require ± 3-degree range of motion, 0.1-arcsec discretization, ~10^6 end-to-end articulations for 10-year mission.

current JPL design concept: linear actuator with DC brushless motor driving a 1mm/rev roller screw

Open issues: Trade-offs between bearing and flexure for central column axis
How to lock during launch (clamp, latch)

POINTS Articulation Actuator Concept



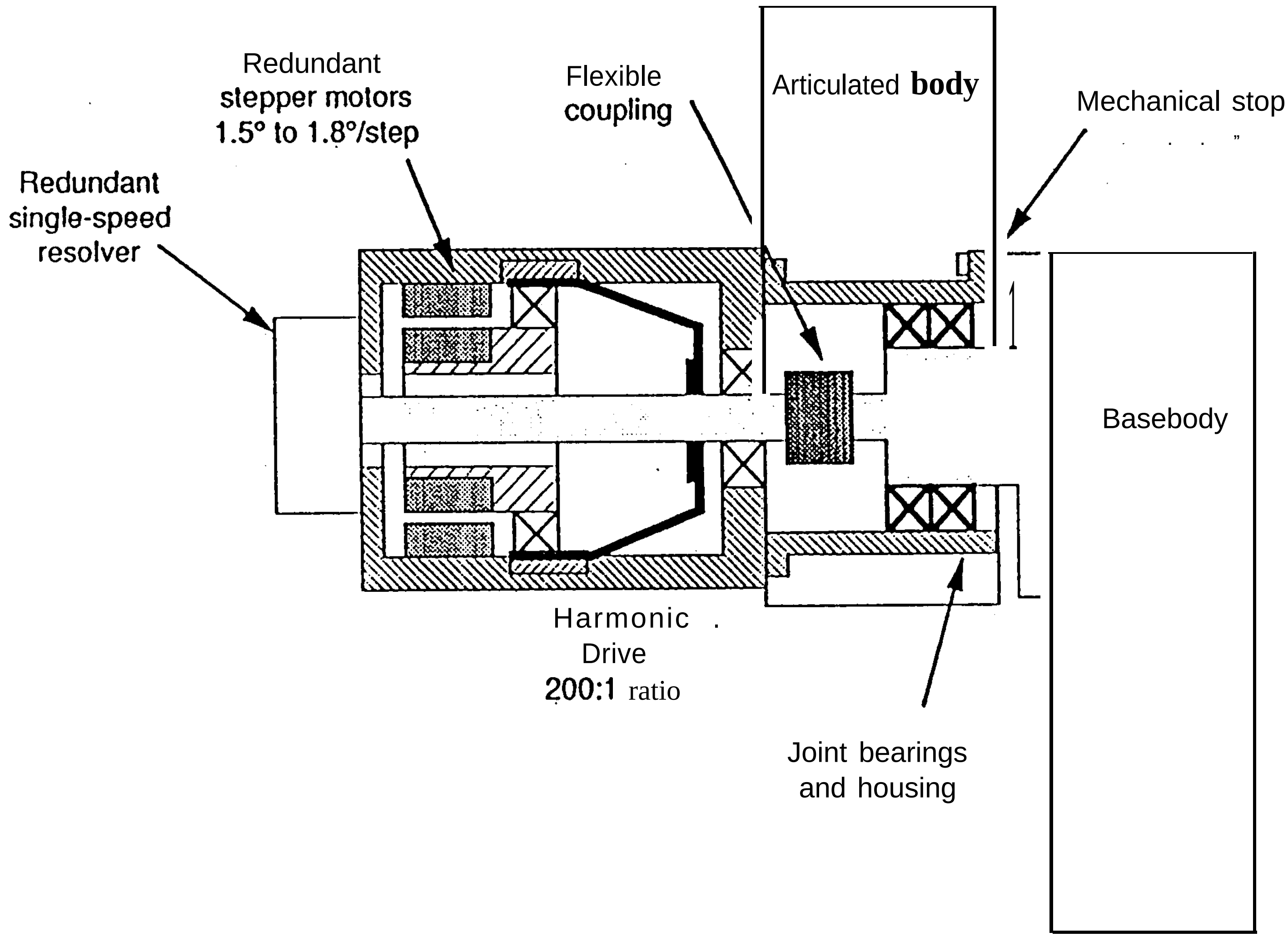
Actuators for POINTS: Gimbal

Require 360-degree range of motion at 1 deg/see, O.1-deg accuracy, lifetime of ‘2x106 radians (5x106 slews averaging 0.3 radians).

Current JPL design concept: stepper motor driving single-stage (e.g., harmonic drive) reducer

Open issues: launch loads, signal transfer across joint (cable wrap problem), disturbance isolation

Gimbal actuator desire concept



Sequence of events for science observations

1. **Adjust relative orientation of interferometers to approximate desired value (90 ± 3 deg)**

   Slew spacecraft (with reaction wheels) to bring target and/or reference stars into star-tracker fields of view (fov)

   Need end-point accuracy ~0.5 deg for tracker with fov ~3 deg x 3 deg (e.g., ASTROS-1, which has 0.3-arcsec accuracy)

   Use gyros to monitor and control slew or count revolutions of reaction wheels (~5 as/rev, read out with 50-mas resolution)

2. **Adjust SC orientation (reaction wheels) and relative interferometer orientation (articulation mechanism), based on information from star trackers and angle metrology system**

   Require l. 0-arcsecond absolute and relative pointing accuracy

3. **Acquire interferometric fringes on brighter of two target stars**

4. **Lock all gimbals and mechanisms**

5. **Integrate on targets**

   Require 3-mas pointing stability over ~ 200 msec

   Use fringe phase measurements to control instrument fine-pointing

Attitude control and momentum management

Attitude-control strategy

3-axis stabilization using 4 reaction wheels, arranged in tetrahedral configuration (or 3 orthogonal plus 1 skewed)

Two star trackers for each interferometer (one prime, one redundant).
Trade-offs to be made between fov and accuracy:
ASTRA (made by HDOS): 7 deg x 9 deg fov, 3-arcsec accuracy;
ASTROS I: 2 deg x 3 deg fox, 0.3-arcsec accuracy

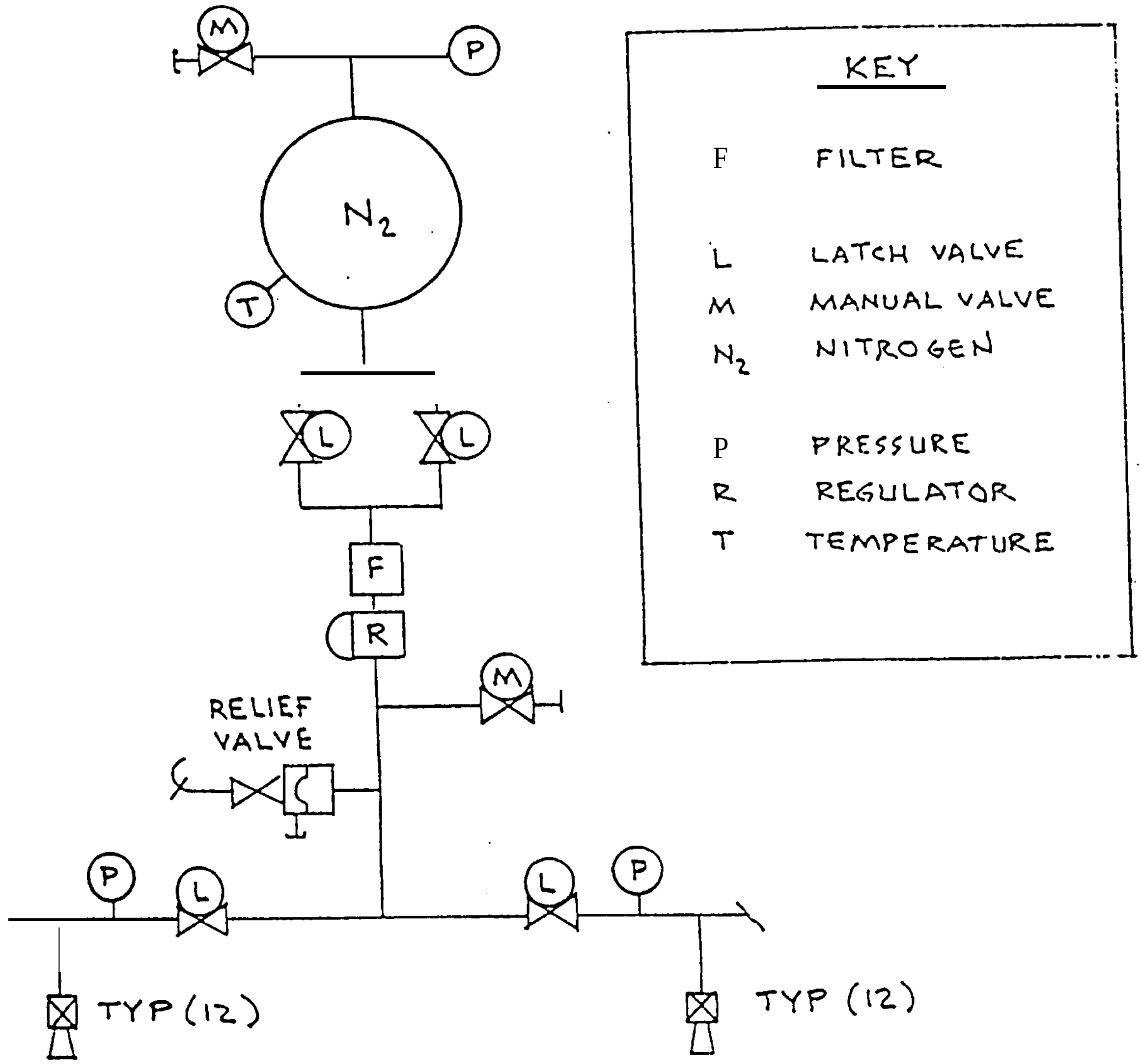
Monitor and control spacecraft slew with gyros, or by counting revolutions of reaction wheels (~260 as/rev, read out with ~3-as resolution)

Momentum-management strategy

High Earth orbit does not permit reliable magnetic torque-unloading

Solar pressure could be used for 2 of 3 axes

Current plan is to use small cold-gas thrusters; estimate ~10 kg of propellant for 10-year mission hl



COLD GAS
GN_2 SYSTEM

FOR POINTS WHEEL DESATURATION

1-6-

RH

Power budget

Telemetry and data storage

Astrophysics-driven telemetry requirements:

Assumptions:

- 200 channels/detector array, 2 arrays per interferometer
- Record photon count from each channel at 1 Hz, with 16-bit number
- Integrate on targets for up to 16 hours per day
- Spend up to 1 hour per day sending data to ground

Results:

- Interferometer data rate ~12.8 kbps; increase to nominal 16 kbps to allow for metrology and other spacecraft/instrument data
- Require 350 Mbytes to store 3 days of interferometer data (Cassini solid-state memory is 500 Mbytes)
- Require downlink data rate of ~256 kbps

Telemetry and data storage

Telemetry options:

S-band downlink to 26-m standard DSN antennas

S- or X-band downlink to n-m DSN antenna being built for space VLBI

Conclusions:

First choice is S-band downlink to 26-m antenna:

50-cm-diameter fixed omni antenna (hemispherical power pattern) with 5 W transmitted power sufficient for 3-dB margin

Second choice is X-band downlink to n-m antenna:
15-cm-diameter fixed omni antenna with 1 W transmitted power sufficient for 3-dB margin

Open issues: **Antenna placement on spacecraft; use of phased arrays**

Telemetry and data storage

Representative link budget: S-band downlink to 26-m DSN antenna

Effect (dB)	180-deg beam	15-deg beam
1O-W SC transmitter (dBW)	10.0	10.0
Transmission losses	- 1.0	- 1.0
Transmitting antenna gain	3.0	**18.8**
Receiving losses (atmosphere, *etc.*)	- 2.0	- 2.0
Receiving antenna gain	52.2	52.2
Propagation losses from 100,000 km	-200.0	-200.0
Boltzmann's constant	228.6	228.6
System temperature (80 K)	- 19.0	- 19.0
Bit rate (256 kbps; dB-bps)	- 54.1	- 54.1
Total E_b/N_0	**17.7**	33.5
Required E_b/N_0 *	11.6	11.6
Margin for 10 W transmitted power	**6.1**	**21.9**

Transmitted power req'd for 3-dB margin: 4.9 W 0.13 w

* QPSK, differential encoding

Telemetry and data storage

Representative link budget: X-band downlink to n-m ground antenna

Effect (dB)	180-deg beam	15-deg beam
1O-W SC transmitter (dBW)	10.0	10.0
Transmission losses .	- 1.0	- 1.0
Transmitting antenna gain	3.0	18.8
Receiving losses (atmosphere, *etc.*)	- 2.0	- 2.0
Receiving antenna gain	58.0	58.0
Propagation losses from 100,000 km	-211.0	-211.0
Boltzmann's constant	228.6	228.6
System temperature (220 K)	- 23.4	- 23.4
Bit rate (256 kbps; dB-bps)	- 54.1	- 54.1
Total E_b/N_0	8.1	23.9
Required E_b/N_0 *	11.6	11.6
Margin for 10 W transmitted power	- **3.5**	**12.3**

Transmitted power req'd for 3-dB margin: 44.7 W **1.2** W

*** QPSK, differential encoding**

POINTS launch sequence

Atlas IIAS launch to *167-km parking orbit

Fire Centaur to inject into elliptical transfer orbit with apogee at 100,000-km

Deploy solar panel.

Alternative options not requiring (full) deployment:

Three-part solar panel (JPL)

Four roof-shaped panels surrounding spacecraft (SAO)

3-axis stabilization during -17-hour cruise to apogee

Spin up to ~20 rpm using small (5-kg) solid-fuel rocket motors

Fire Star-37FM solid-fuel rocket motor to circularize orbit

Spin down with small (5-kg) solid-fuel rocket motors

Complete deployment of solar panel(s); stabilize spacecraft

POINTS launch-sequence considerations

Solid-fuel rocket motor chosen for orbit circularization primarily because of difficulty maintaining liquid fuel during 17-hour cruise to apogee

Combination of Atlas IIAS and Star-37FM can place spacecraft mass up to ~1700 kg in 100,000-km orbit at 28.5-deg inclination (to ecliptic)

Current spacecraft mass estimate is ~1390 kg (JPL FY92 configuration)

Higher inclination preferable to minimize Earth-shadowing, but costs in mass; cost is less than 30 kg to change inclination by less than 15 deg.

Any orbit well above radiation belts is acceptable from standpoints of science, telemetry, and orbit determination.

Could add ~100 kg mass by reducing apogee to 80,000 km

Performance envelopes for Centaur, Star-37FM, and attitude. control subsystem guarantee orbit well within 90,000 - 110,000-km range, even for 3-σ errors in all parameters

Some solar power necessary during long cruise to HEO. Must assess ability of solar panel to withstand Star-37FM burn (3.5 g maximum)

POINTS orbit determination

Require velocity-determination accuracy of -0.6 mm/s to reduce contribution of stellar aberration to astrometric measurement error to below 0.4 μas

Three possible strategies considered:

1. **GPS receiver on POINTS spacecraft**
2. **GPS-like beacon on POINTS spacecraft**
3. **Traditional Doppler tracking**

Option 2 (GPS-like beacon on spacecraft) meets requirements easily

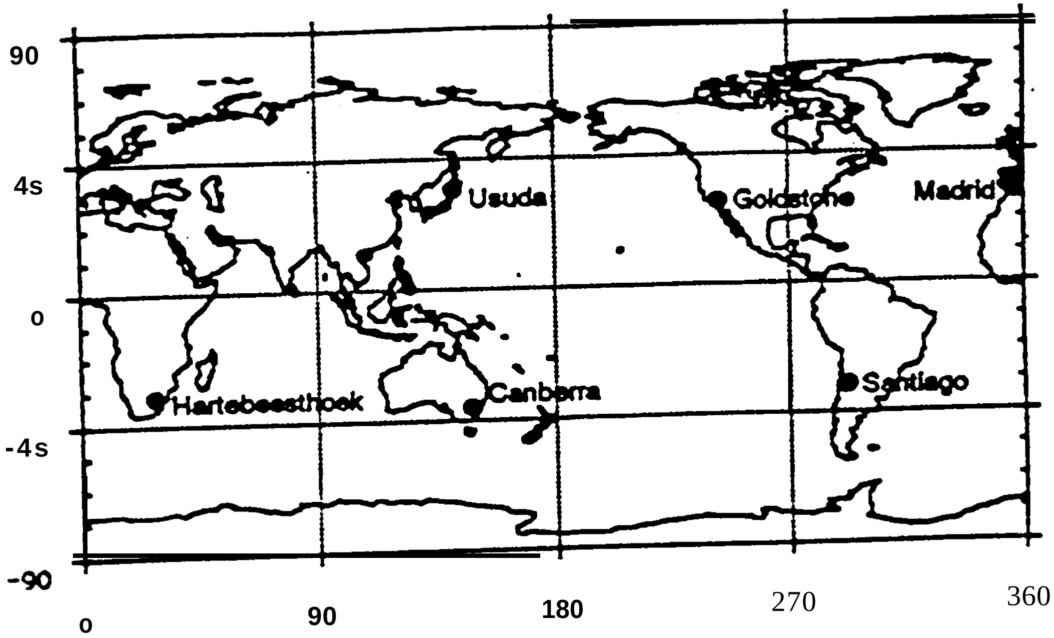
Beacon would have 2 tones at L-band (1.2-1.6 GHz), or 1 tone plus sidebands at X- or Ku-band (8 or 15 GHz)

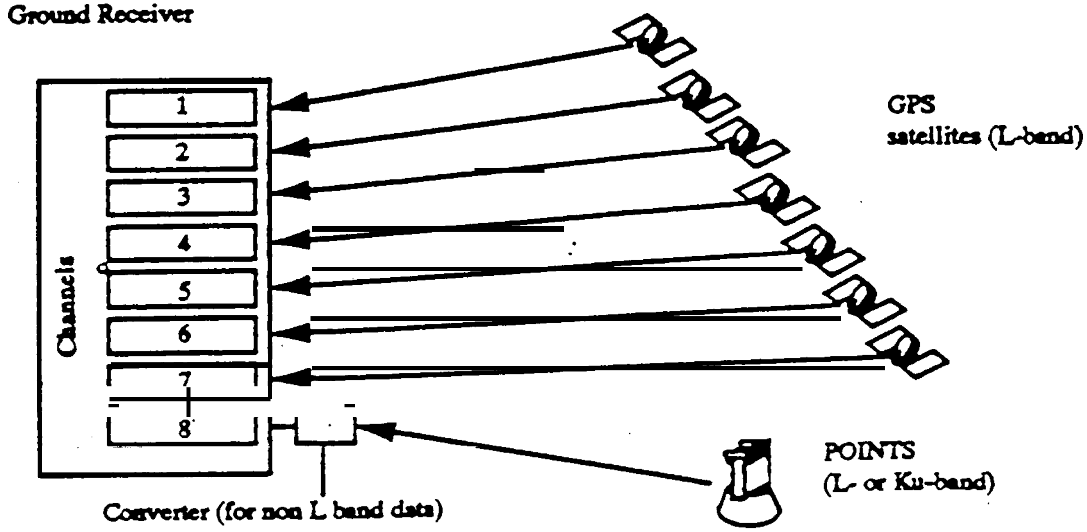
Track spacecraft with 8-channel GPS receivers located around the world

Each receiver tracks GPS satellites in 7 channels, with intermittent tracking of POINTS in 8th channel

The six TOPEX sites would be adequate

POINTS orbit determination





POINTS orbit determination

Covariance analysis results for use of GPS-like beacon on spacecraft:

Velocity error of 0.1 mm/s achievable by tracking for 2 of every 8 hours

Orbit can be predicted forward for > 6 days and still meet 0.6 mm/s req't

Expect 0.6 mm/s accuracy achievable with < (<<?) 10% duty cycle

Spacecraft hardware:

L-band or Ku-band options are feasible for spacecraft by broadcasting a low-power signal through a pair of switched omni antennas (one on Sun-facing side, one on back)

L-band:	**15 W (DC), 5.5 kg, 5900 cm3, $\Delta f/f^{*} \approx$ 10-10**
Ku-band:	**22 W (DC), 3.6 kg, 8200 cm3, $\Delta f/f \approx$ 10-11**

Ground hardware:

L-band system preferred because standard GPS channels can be used

Frequency allocation at Ku-band may be easier to obtain

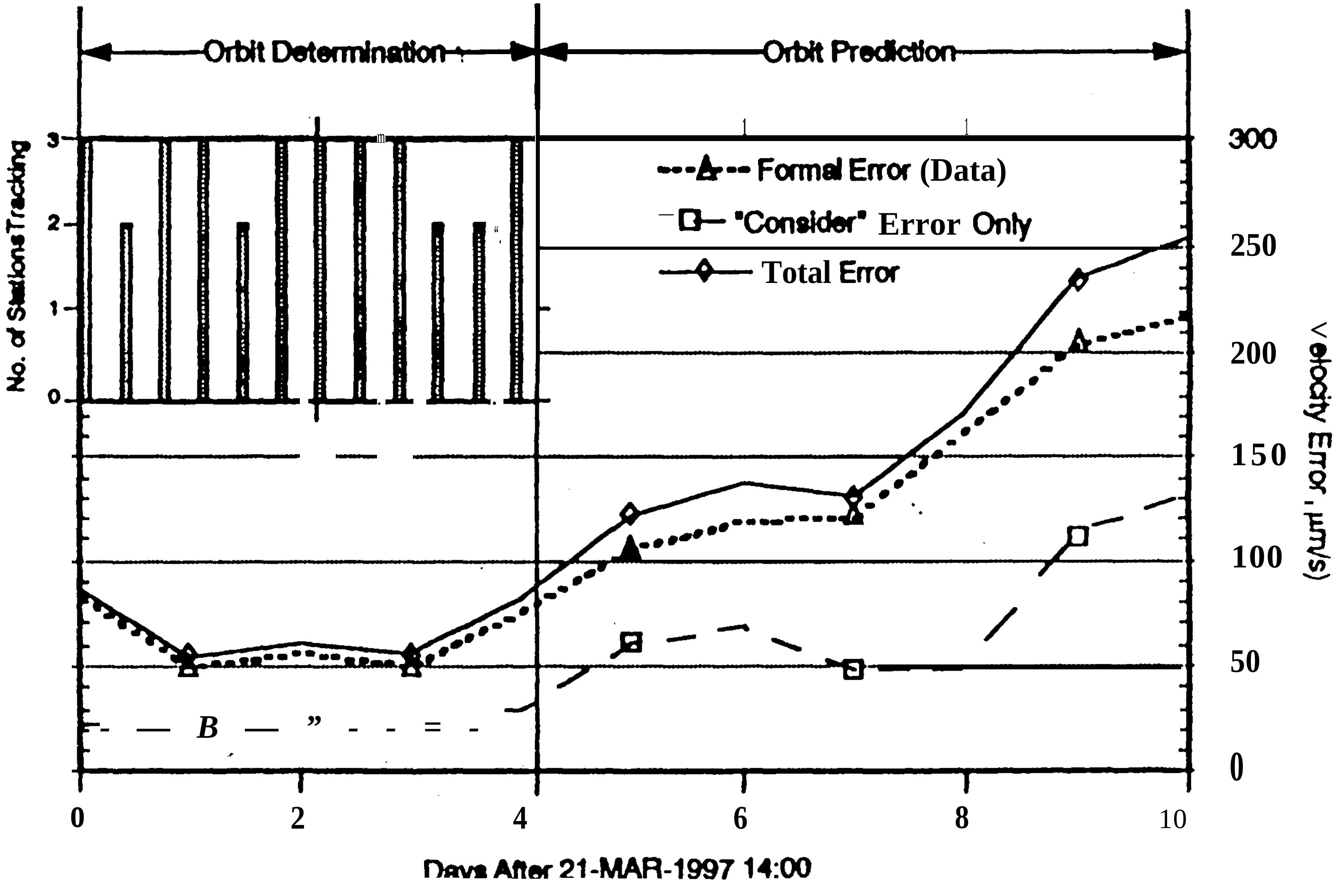
*** Fractional frequency stability**

POINTS orbit determination

Ku-band; track for 2 of every 8 hours during first 4-day orbit, then predict orbit for 6 days.

Use 6-station TOPEX GPS network, with continuous GPS tracking.

"consider" parameters: gravity field, UT, polar motion, troposphere, ionosphere



Precision Actuators for Spaceborne Interferometers: A Tutorial

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ADSTRACT

There seems to be a strong correlation between the number of moving parts on a spacecraft, and the quality and quantity of science that it can be achieve, This is especially true for applications with demanding pointing and alignment requirements like spaceborne interferometry Unfortunately, moving parts are expensive, and the destre to add moving parts to maximize science conflicts with NASA's current climate of costs constraints. The intent of this paper is to provide the interferometer (or other mission) designer with an overview of the technical issues that confront the cost-effective design and specification of precision spacecraft actuators.

First, the paper describes the capabilities and limitations of common actuator components such as bearings, prime movers, and displacement sensors. Next, the paper describes some generic actuator configurations for typical applications. Finally, the paper provides Ups on how to write actuator requirements,

1. INTRODUCTION

Proposals for spaceborne interferometers, such as the Precision Optical INTerferometer in Space (POINTS) and the Orbiting Stellar Interferometer (0S1), come at a particularly challenging time. Cost constraints are tighter than ever, and a string of mechanical failures on NASA spacecraft has fueled skepticism of "big science." One effect of the emerging "smaller, cheaper, better" philosophy of spacecraft design has been to eliminate subsystems which most visibly complicate the spacecraft--in particular, moving parts.

Unfortunately for space interferometry, moving parts--in fact, some of the most precise actuator systems ever proposed--are integral to the instrument's design. It will be the spacecraft engineer's challenge to design a system which can do the job with the simplest, most reliable, and least expensive actuators possible, The goal of this tutorial is to sensitize the reader to the issues involved in specifying, designtng, and using precision space actuators.

2. ACTUATOR COMPONENTS

Key to understandtng the constraints and challenges of designing and specifying flight actuators is a thorough understanding of actuator components. The main components, shown in figure 1, are the bearings, prime mover, transmission, sensor, and signal transfer device. A guide to understanding these components follows:

2.1. Bearings

The job of a bearing is to *predictably allow* motion in a particular axis or axes, while constraining motion in the other axes. Bearings defined in this way include sliding-element bearings, rolling-element bearings, flexures, and magnetic suspension.

2.1.1. SlidinQ-element bearings

Sliding-element bearings are seldom used in precision space applications because of the difficulty in maintaining lubricity, and the typically high friction. Common materials are hardened steel for the shaft, rotating in a soft sleeve. Sleeve materials can range from oil-impregnated porous bronze to polymers such as PTFE (Teflon) or Vespel. Sliding bearings require either tightly-controlled clearances or grooves where wear debris can accumulate without jamming the device. As wear progresses, the clearance will increase, diminishing the bearing's precision. A preload (typically applied with a spring with enough travel to cover the expected wear) can keep the clearance closed, but then the bearing's stiffness is limited to the preload stiffness.

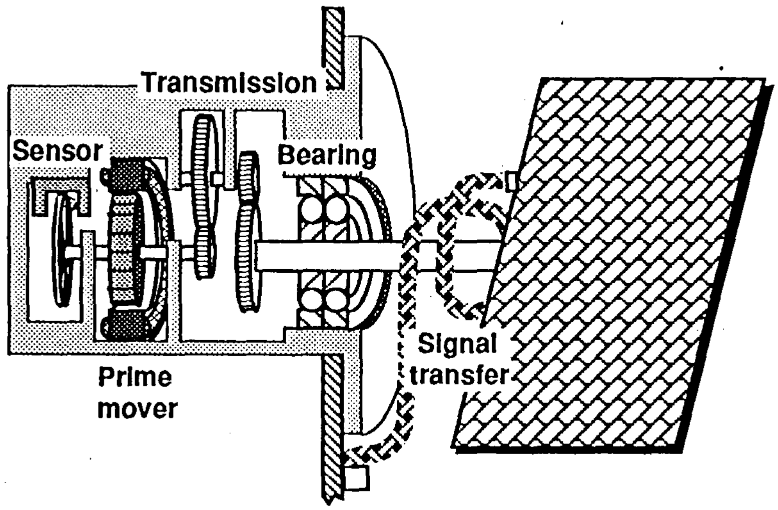

Figure 1, Major components of an actuator.

2.1,2, Angular contact ball bearings

The most common bearing device is the angular contact ball bearing, shown in figure 2. Angular-contact bearings are preferred because they exhibit less friction than roller or needle bearings, and most importantly, their structural stiffness in the non-rotating axes can be very well modeled and controlled.

The fundamentals of bearing configuration are illustrated in figure 2. The angular contact bearing allows the contact angle to be specified, (typically between 30° and 4 5°), and in this way the cross-axis stiffness can be controlled, A single angular contact bearing can only support thrust in one direction, therefore they are always used in pairs. The pair are pre-loaded against each other, such that a thrust load must be greater than the preload before the bearing will separate. Bearing race widths are ground such that when a pair are pressed against each other (or against a spacer) to the point that the races come into contact, the bearing will be under a known preload. This is called a duplex pair. For extremely low preloads, or cases where thermal expansion of the actuator housing can change the bearing spacing, and thus the preload, bearings can be loaded with a spring, typically in the form of a "wavy washer." The spring stiffness must be low enough to generate a relatively constant load over the expected range of displacement.

High preload Is necessary to resist high loads (such as launch conditions), and bearings are sized for that particular worst load case. Unfortunately, that same preload increases friction; thus load and performance requirements on an actuator are always in conflict. Confounding this conflict is the problem of predicting friction while at the specification stage of a project. Existing models are not particularly accurate. To solve this problem, TRW has proposed a mechanism to actively control bearing preload, relaxing it after launch to reduce operating friction [1].

The right-hand figure illustrates a bearing pair arranged "back-to-back" or "DB". The lines of force converge away from each other, making this the stiffest arrangement in cross-axis moment loading, and necessary if the bearing patr can not be physically spaced far apart. The center figure illustrates a pair arranged "face-to-face" or "DF," Here, the lines of force converge toward each other. The bearings can be spaced to minimize cross-axis moment stiffness, which may be desirable to not cinematically overconstrain a mechanism,

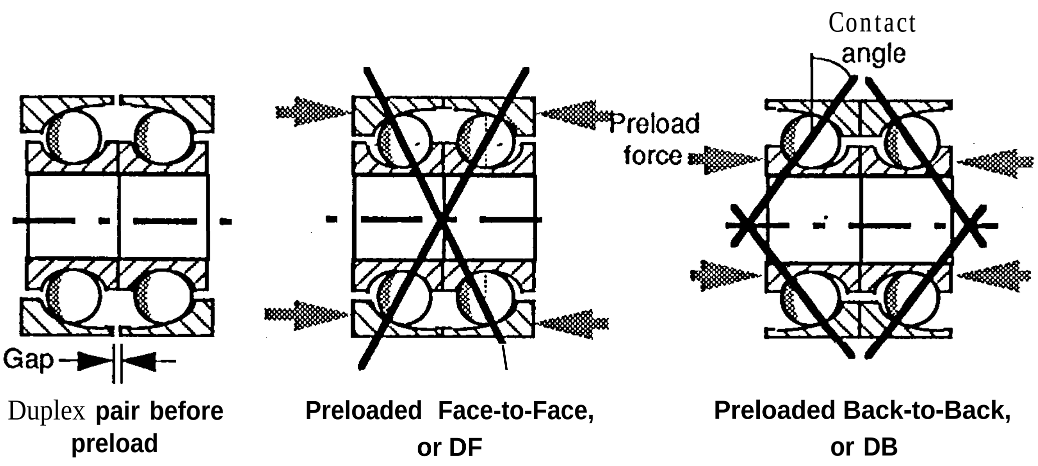

Figure 2. Angular contact bearing configurations.

Models and vendor data exist which permit a fairly accurate prediction of bearing stiffness in the non-rotating directions. However, predicting friction about the rotating axis is another story. Computer programs exist which can predict coulomb and viscous friction levels based on bearing configuration, load, and lubrication, but in our experience these programs are barely accurate to within an order of magnitude for the low rotation rates common to spacecraft applications (<100 rpm). The programs are useful only for comparing design options. For precision pointing applications, it is critical to be able to model the friction discontinuity that occurs when a bearing's direction is reversed. In lubricated ball bearings, "stiction", a higher break-away friction than running friction, is almost never observed. Instead, bearing friction at rate reversal follows a hysteretic curve as shown in figure 3b. This curve is described by the Dahl mode12, which incorporates a hysteretic initial sttffness that approaches the bearing coulomb friction level. Some preliminary work has been done to predict Dahl friction parameters as a function of bearing geometry[3], but in practice these parameters must be experimentally determined. Until more research is performed, bearing friction, and therefore pointing performance, can not be well-predicted until the bearing has been installed and tested,

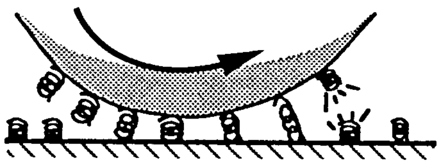

Figure 3a. The adhesion theory friction.

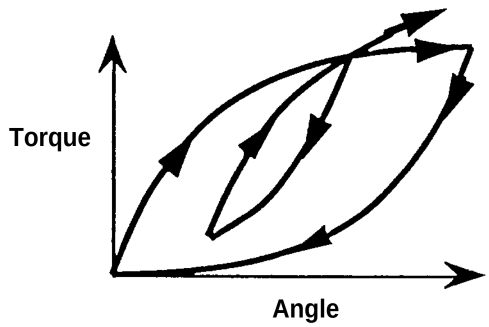

Figure 3b. Typical hysteresis loop as described by the Dahl friction model.

Bearing balls and races are most often made of hard, corrosion-resistant ferrous alloys such as 440C stainless steel. Ceramic (silicon nitride), or ceramic-coated (titanium carbide) balls have also been used recently. The ceramic's high stiffness reduces the contact area, reducing friction somewhat, and life is apparently increased because the ceramic does not micro-weld with the steel race[4]. However, there is little flight history with ceramic bearing materials,

Bearing quality is defined by what is called an ABEC (Annular Bearing Engineers' Committee) tolerance. The ABEC tolerance is a scale from 1 to 9, and describes the variation in bearing tolerances, in particular the variation in location of the axis of rotation (this is called runout). Bearings used in flight applications are typically ABEC 7 or higher.

Bearing life is traditionally predicted by fatigue life. Bearing vendors will refer to the "L1 O" life, the time in which the probability of fatigue failure reaches 10%. Bearing analysis programs such as BASDREL (written by AVCON of Northridge, CA) can calculate L1O life. The failure criteria for bearings in space mechanisms is more difficult to define. A precision actuator bearing has failed if the torque required for rotation exceeds the actuator's torque margin, or if bearing torque noise has exceeded the limits which allow acceptable smoothness of motion. This type of failure is usually due to lubricant starvation. The failure criteria for a particular application is so unique that analytical predictions of life are useful only in comparat ive design evaluations. The only accurate way to predict life is with a life test, Because the effects of bearing speed and temperature on bearing life can not be accurately analytically described, accelerated life tests are not practicable. This means that proving 10 year life for a reaction wheel requires a 10 year life test (preferably with a large sample of wheels). In general, the best way to build confidence in reliability is by comparison with other long-lived devices.

The point of a lubricant is to separate two high-shear-strength sliding surfaces with a low-shear-strength material, According to the adhesion theory of friction[5], at low or zero speed, asperities on a bearing ball (shown as springs in figure 3a) weld to the asperities on a bearing race. As the ball begins to roll and slide, the asperities stretch (causing the spring-like initial slope in the Dahl friction model) until the welds break (at the plateau of the Dahl curve). Using extremely smooth materials reduces the number of asperities, and thus the friction. Using a ball material different from the race material (ceramic, for instance) can also reduce or eliminate welding, as can certain race coatings.

Good lubricants have low shear strength (low viscosity), and high surface tension so they will wick between a ball and race. For space applications, the lubricant should also "have low vapor pressure so as not to evaporate into the vacuum of space. Viscosity should be insensitive to the expected temperature range for predictable performance. Lubricants also should not chemically react with the rest of the bearing. Solid lubricants such as MoS2 consist of plates that slide over each other such that the effective shear strength is extremely low. Unfortunately, solid lubricants have no ability to wick back into a bearing, so eventually they wear away and the bearing is left unlubricated. Liquid lubricants are most effective as the bearing picks up speed. A ball will tend to hydroplane over a liquid-lubricated surface, and its asperities will no longer touch those of the race, When the ball is completely supported by a layer of fluid, this is called hydrodynamic lubrication. Over time, a lubricant degrades because of contamination with wear particles from the bearing, or from increasing viscosity as the less-viscous fractions evaporate, or from chemical degradation. To increase bearing life, the lubricant must be replenished or at least the loss rate must be minimized. In general, the oil initially put into a bearing in a space application is all the bearing will have throughout its life. Ball retainers can be made of porous materials that are impregnated with lubricant: over time, some of the lubricant wicks out and is transferred to the balls by contact. Bearings can also be packed with grease. The grease doesn't act as a lubricant but rather as a an oil reservoir; oil in the race is pushed aside into the surrounding grease as a ball passes by, but then wicks back out of the grease, onto the race. Another technique is to place a sacrificial reservoir of lubricant (such as an impregnated porous material) in the vicinity of the bearing. Oil will evaporate from the reservoir, raising the local vapor pressure and reducing the rate of evaporation from the bearing. Honeywell has proposed a system to actively resupply lubricant to a bearing[6], but it has not yet flown,

2,1,3, Flexures

A simpler, more predictable means of constraining motion is with a flexure. A flexure is simply a spring or group of springs designed to be much stiffer in some degrees of freedom than others, For rotary applications, a common configuration is the cross-flexure, also called the flex-pivot, shown in figure 4a. The flexure's stiffness in all degrees of freedom can be accurately calculated using beam equations. To first order, the flexure can be modeled as a torsional spring with constant stiffness. In fact, the torsional stiffness changes somewhat with angle and load (increasing under compressive load, decreasing under tensile load[7]), and the axis of rotation also moves with angle. These second-order effects can also be analyzed using beam equations, For more precise rotary applications, the triflex pivot (figure 4b) can be used. The third flexure further constrains the motion of the axis, provides greater stiffness, and allows the flexure to be preloaded in tension to reduce torsional stiffness. Cross-flex pivots are commercially available, primarily from Lucas Aerospace of Utica, NY. 'hi-flex pivots are typically custom-made.

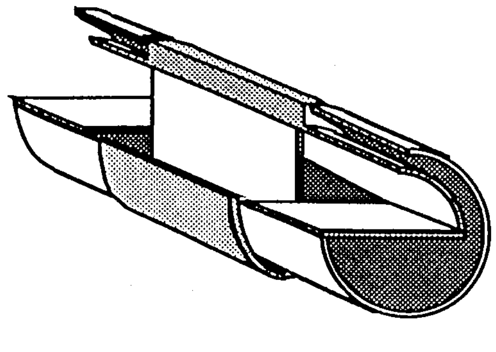

Figure 4a. Cutaway of the Lucas Aerospace Freeflex pivot.

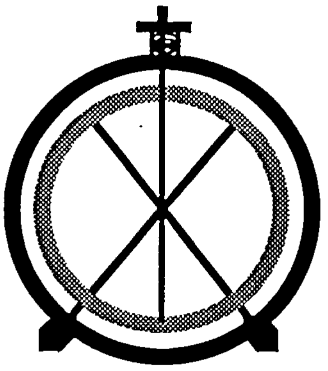

Figure 4b. End view of a pre-loaded tri-flex pivot.

The torsional stiffness of a flexure may lead to higher torsional loads than for a ball bearing, but the ability to reliably predict that load and the lack of torque discontinuities makes the flexure preferable for applications requiring smooth motion. When the limited range of motion of a flexure can not be tolerated, two-stage devices can be built which use the flexure stage only for fine pointing and compensation of ball bearing torque disturbances, Also, the fatigue life of the flexure can be predicted or the flexure can be designed for indefinite life.

There are a variety of other flexure configurations for various desired motions; the same design techniques and design rules used in kinematic optical mounts can be applied to flexure design for mechanisms.

2. 1.4. Magnetic bearings

The "technology of the future" that solves all the drawbacks of ball bearings (torque discontinuities, limited life) and flexures (limited range of motion) is the magnetic suspension. The suspended object completes the flux path between two electromagnets. The object is at unstable equilibrium, so a cent rol loop using position feedback, usually from eddy-current sensors or capacitive sensors, stabilizes the suspension, The precision of the suspension depends on the accuracy of the sensor (which can exceed ball bearing accuracy) and the stiffness depends on the control loop bandwidth, Stiffness and load capacity can be comparable to that of ball bearings, although suspension bandwidth is typically limited to below a few hundred Hertz.

Magnetic bearing technology is fairly mature. Several systems are commercially available from companies including AVCON of Northridge, CA and SatCon Technology Corp. of Cambridge, MA. They have been used in reaction wheels in the ESA SPOT satellite. The drawback of a magnetic bearing is the additional cost, mass, and power of the electronics. Magnetic suspensions have not yet flown on any JPL spacecraft.

2.1.5. Bearing comparison

The capabilities and constraints of the three types of bearings discussed above are summarized in the following table:

	Angular Contact Ball Bearing	Flexure	Magnetic Suspension
Range of motion	Continuous	<±250 mrads	Continuous
Stiffness of constrained axes	Highest, predictable	Predictable	High but bandwidth-limited, predictable
Axis of rotation precision	Runout as small as 2.5 μm	Moves with rotation; >0. 1% of diameter at 170 mrad deflection	Equivalent to ball bearing
Friction, torsional stiffness	Dahl friction, difficult to predict	Predictable torsional stiffness, increases with load capacity.	Virtually zero friction and torsional stiffness
Life	Prediction based on previous experience	Can be designed for infinite life	Limited by electronics Onlv
Temperature range	Limited	Widest	Wide
Contamination	Lubricant must be contained	None	None
Availability	Wide variety of sizes, configurations	Generally requires custom design	Limited flight heritage

Table 1. Bearing Comparison

2.2 Prime movers

Classes of prime movers include motors that work on the Lorentz force, those that work by minimizing the reluctance path of a magnetic field, and those that utilize "smart materials", whose dimensions change with applied electric or magnetic filed, or temperature. The discussion here will be limited to electromagnetic motors.

2.2.1 Voice coil

The simplest prime mover driven by the Lorentz force is the "voice coil" linear motor. It consists of a wire bobbin whose windings cross a permanent magnetic field. Force is generated according to well-known equation; $\vec{F} = \vec{I} \times \vec{B}$. Force is linear with current and independent of stroke (as long as the winding is within the field) which makes it an ideal prime mover for isolation systems. Stroke is limited to the width of the magnetic field (usually no more than a few centimeters or so) and the device provides no motion constraint in other directions; it requires a linear-motion bearing.

2.2.2. Brushless motor

There is a rotary equivalent of the voice coil, called a limited angle torquer. This device operates over a limited angle, generally less than 120°, and generates torque linearly proportional to current. For continuous rotation, the motor requires multiple windings, and current has to be switched from winding to winding such that the energized winding is the one most orthogonal to the magnetic flux (and capable of generating the most torque). This is called commutation. In conventional DC motors, commutation is accomplished with brushes which mechanically switch windings on and off as the windings rotate. For precision space applications, brush motors are undesirable because of the friction, difficulty of predicting brush wear in vacuum, and concern over conductive brush debris floating around in O g, causing short circuits or contaminating the mechanism. Instead, brushless motors are used,

A brushless motor is shown in figure 5a. The brushless motor is commutated electronically, using an angle sensor to determine which winding to energize. Unlike the limited angle torquer, whose torque is nearly constant with angle over its operating range, torque from a single winding, or phase, of a brushless motor is nearly sinusoidal with angle. The number of sinusoids per revolution depends on the number of magnet poles in the motor; the figure shows one pole, but three or more are commonly used,

The simplest way to commutate a brushless motor is to switch the current (and current direction) on and off as shown in figure 5b; this is called square wave commutation, and is equivalent to how a brush motor operates. Torque is approximately proportional to current, but has bumps called "torque ripple. " The magnitude of the ripple is proportional to the commanded torque, and its frequency equals twice the number of poles times the number of phases. The advantage of square wave commutation is that it requires only coarse angle knowledge. Sensing is typically accomplished with a Hall effect sensor, a solid-state magnetic field sensing device. One Hall sensor per phase is located near the motor rotor and senses the magnet poles as they pass by. Commercial brushless motors generally come with Hall sensors built-in.

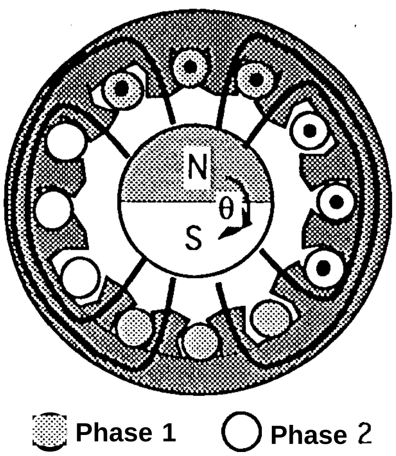

Figure 5a. A DC brushless motor

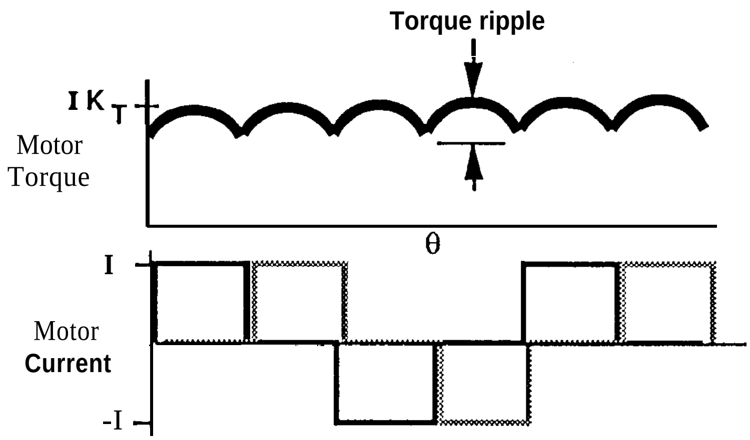

Figure 5b. Square-wave commutation and torque ripple.

If necessary for smoother operation, torque ripple can be minimized by using sinusoidal commutation. The current into each motor phase is multiplied by the sine or cosine of the rotor angle, based on feedback from a high-resolution angle sensor. The sinusoidal variation in torque is canceled out, and the motor produces torque nearly independent of angle. obviously, sinusoidal commutation requires much more complicated electronics than square wave commutation, and is therefore more expensive.

Another source of torque disturbance in DC motors is called cogging. Cogging is caused by the rotor poles wanting to line up with the iron slots around which the windings are wound. Cogging is a roughly sinusoidal disturbance, with frequency determined by the number of slots and the number of poles. Cogging can be eliminated by not using iron slots to support the windings, but motor efficiency is reduced because the magnetic field is less intense.

The DC motor is inherently a device for controlling torque. The ability of a sinusoidally-commutated motor to exert torque independent of angle makes it ideal for applications requiring smooth motion, or where the controlled body needs to be isolated from the motion of the base body, such as in inertial pointing.

2.2.3. Step motors

Another type of motor commonly used in precision spacecraft applications is the step motor, It operates on the principal of minimizing the reluctance of the magnetic flux path between the rotor and stator. The step motor has a permanent magnet rotor, like a DC motor, but the magnetic fields from the windings in the stator line up with, rather than cross, the fields from the rotor. This is shown in figure 6. As each winding is energized, the rotor steps to align itself with that winding.

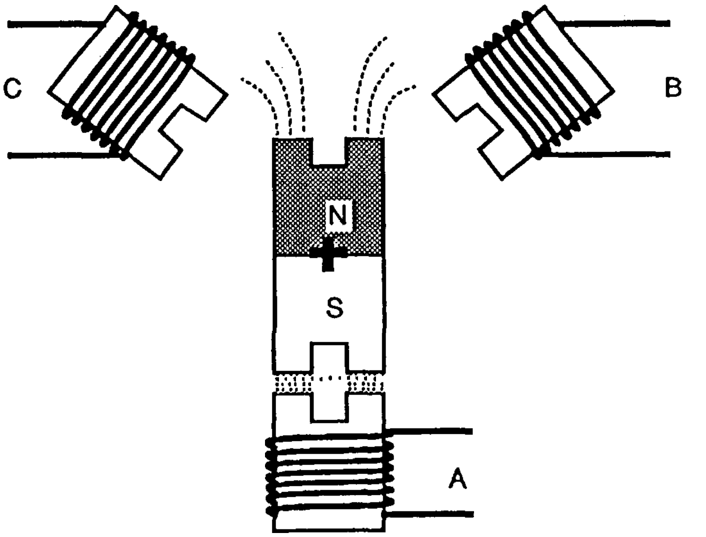

Figure 6. Schematic diagram of a stepper motor.

As shown in the figure, the rotor and stator have "teeth" which concentrate the magnetic field at particular angles so that the steps are more precisely defined (the reluctance of the flux path is minimized when the teeth are aligned). When the windings are turned off, the rotor's magnetic field will cause it to stay aligned with the nearest teeth, thus a stepper motor has unpowered holding torque.

Motor direction is controlled by pulsing the windings in a particular sequence, and average speed is controlled by controlling the sequencing rate (although the instantaneous speed varies from zero at the beginning and end of each step, to a peak in the middle of the step). Torque can be controlled by increasing the current, or by increasing the pulse duration. In practice, stepper drive electronics use a fixed current and pulse width, and vary only the time between pulses for rate control. For a winding energized with constant current, the torque exerted on the rotor varies from a maximum when the rotor is a full step away from the energized winding, to zero when the rotor is aligned.

Steppers come in a variety of configurations. typically having from two to six phases and step sizes from 90° to 1.5°, Even finer motion can be achieved by microstepping. Microstepping is achieved by energizing two adjacent windings at a time. By varying the ratio of current between the two windings, the equilibrium position for the rotor can be varied. Of course, when the motor is turned off, the rotor will align itself with the nearest mechanical pole. Like a conventional stepper, microstepping can be accomplished without position feedback (at the risk of "slipping" steps under high loads), but the electronics are slightly more complicated than for normal stepping. Commercial microsteppers are available with as many as 50,000 steps per revolution.

2.2.4. Motor Comparison

A comparison of the features of DC brushless and step motors is summarized in the table below:

	Brushless motor	**Stepper motor**
Motion increment	Continuous	1.6 rads to 26 mrads per mechanical step, 125 μrad per microstep
Power efficiency	High	**Low**
Holding torque	Requires power	Passive detents at mechanical steps
Rate stability	Smooth	Inherently poor
Torque modelability	Easy to model	Difficult to model
Open-loop operation	**No**	Yes
Electronic Complexity	Complex	Simple

Table 2. Motor comparison

2.3 **Transmissions**

Entire books have been written describing all the mechanical transmission devices available to a designer, but this paper will limit itself to the discussion of devices commonly used for precision space mechanisms: spur gears, harmonic drives, bands drives, and ball and roller screws.

2.3.1. Spur gears

Spur gears are the most common type of gear used because they can be manufactured with great precision, and the higher load capacity of helical gears is generally not required for precision pointing applications. The teeth on a spur gear are involute-shaped, so that the meshing teeth roll rather than slide against each other. Clearance is required between meshing teeth to allow thermal growth and machining tolerances, therefore there is lost motion, called backlash, when the direction of a gear is reversed, Backlash is the largest error associated with spur gears, although there is also motion error due to dimensional imperfections. Backlash can be eliminated through the use of "anti-backlash" gears, illustrated in figure 7. Anti-backlash gears are actually two gears on the same shaft preloaded against each other by a spring to take out the clearance in the mesh.

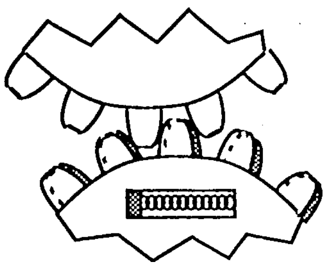

Figure 7, The anti-backlash gear.

Good design practice limits the gear ratio in a single pass to no more than 6:1 8 Nonetheless, actuators on the Voyager spacecraft achieved a final ratio of 9000:1 using six passes. A more compact technique to achieve high ratios is the planetary arrangement, such as the transmission used in a three-speed bicycle. Ratios of up to 100:1 can be achieved. The planetary arrangement also supports greater loads because three gears share the load

2.3.2. Harmonic Drive

The Harmonic Drive can provide very high mechanical advantage with a single mechanism. Thus, the Harmonic Drive is simpler, and arguably more reliable than a spur gear train with the same gear ratio. Figure 8 illustrates how a harmonic drive works. It consists of three parts: An elliptical "wave generator," surrounded by an elliptical ball bearing, sits inside a cup-shaped "flex-spline." The flex-spline has external teeth that mesh with a "circle spline," which has internal teeth. A typical configuration for speed reduction is to drive the wave generator, fix the circle spline, and take power off of the flex-spline. The flex-spline has two fewer teeth than the circle spline. Assume the circle spline has N teeth; when the wave generator has made half a revolution, as shown in the figure, it has caused N/2 teeth to mesh. Since the flex-spline has fewer teeth, it will have rotated one tooth in the opposite direction of the wave generator, for a gear ratio of-N: 1. Gear ratios are available ranging from 60:1 to 200:1. Harmonic Drives are available from the Emhart Machine Group of Wakefield, MA.

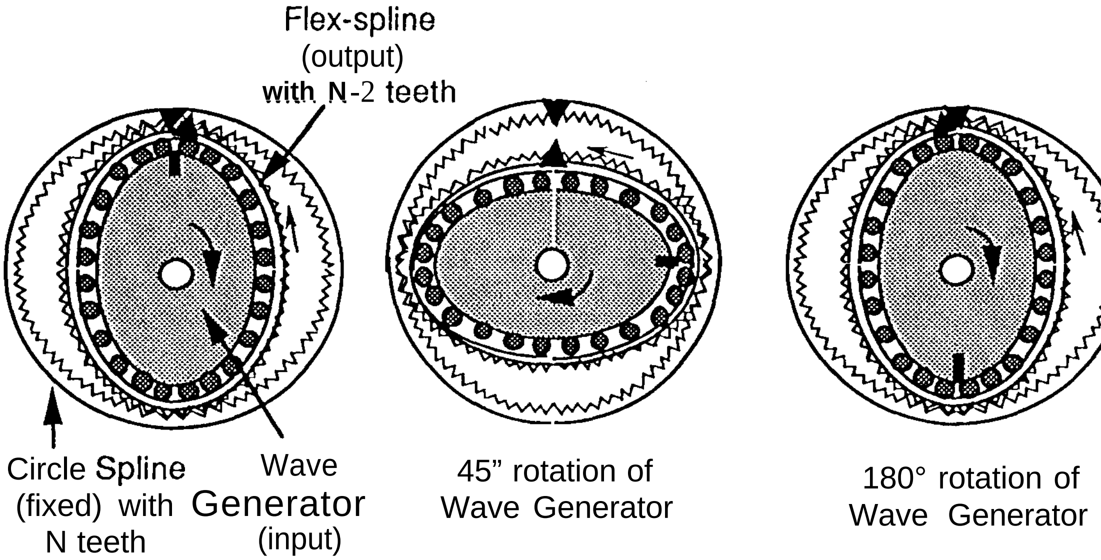

Figure 8. Harmonic Drive operation.

The meshing motion in a harmonic drive is different from that in spur gears, so harmonic drive teeth are nearly triangular, and exhibit no backlash. However, the harmonic drive does have a larger gear error than an equivalent spur gear train. The error is due primarily to dimensional inaccuracies. The error has a frequency of twice per wave generator revolution, with an amplitude modulation of twice per flex-spline revolution. The amplitude of the error can be large enough to cause the output to rotate backwards over small angles, therefore the harmonic drive should not be used when great precision or rate stability is required.

2.3.3. Band drive

A more precise transmission is the band drive, Bands material is typically a high-tensile strength, high fatigue life alloy such as Elgiloy, and bands are preloaded in tension to maximize the stiffness of the transmission. The radial bearing load, and thus bearing friction, can be decreased by using crossed bands; the tension in each band acts tangentially on the rotating members, and the two tension vectors cancel each other out. This configuration isshown in figure 9. A band drive of this type has less friction than a mechanism with meshing gear teeth, and can be manufactured to tighter tolerances than a geared device. The achievable mechanical advantage is on the order of less than 10:1. The band drive can also be configured analogously to a rack and pinion in order to convert rotary to linear motion. The major advantage of the band drive is that it is the cleanest transmission in terms of torque disturbances, making it ideal for applications which require high rate stability, The major drawback of this type of band drive is that motion is limited to a fraction-of a revolution.

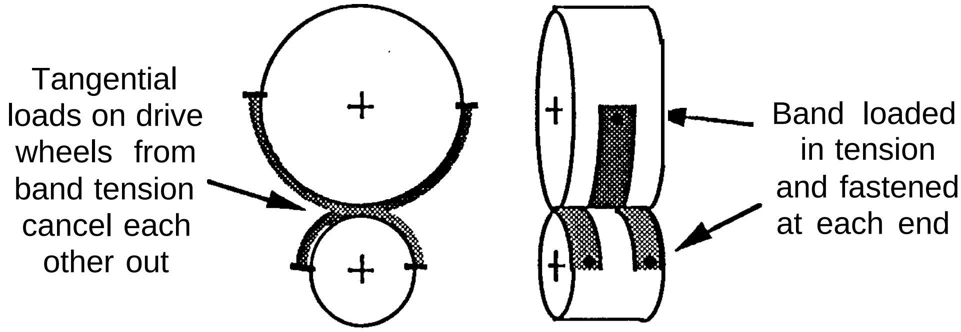

Figure 9. The split band drive.

2.3.4. Bal land roller screws

A commonly-used precision transmission for converting rotary motion to linear motion is the ball screw. This device, shown in figure 10a, consists of a threaded nut and shaft with balls between the two to reduce friction. Balls in the ball-screw recirculate; that is, when the balls advance to one end of the nut, they enter a tube wrapped around the nut which directs them back to the other end. The threads are cut with a curvature that conforms to the ball, such that when the ball screw is loaded, the load is transmitted through the balls at some contact angle as in an angular contact bearing. Ordinarily, there is some backlash due to the clearance between the balls and the threads, but nuts are available in preloaded pairs with no backlash (sometimes as two independent loops of balls in the same nut). Ball screws exhibit friction analogous to that in ball bearings, plus they generate torque disturbances due to the balls clicking in and out of the recirculation tube. They also exhibit positioning errors (the error between ideal and actual stroke for a given nut rotation) due to dimensional errors in the thread spacing on the screw. They are available in a wide variety of thread pitches and load-carrying abilities.

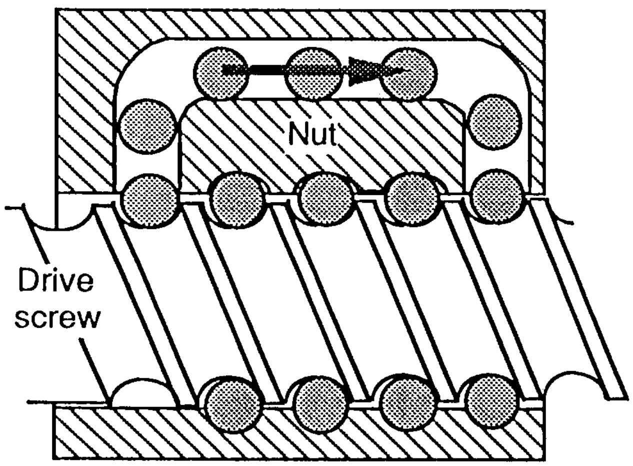

Figure 10a, A ball screw

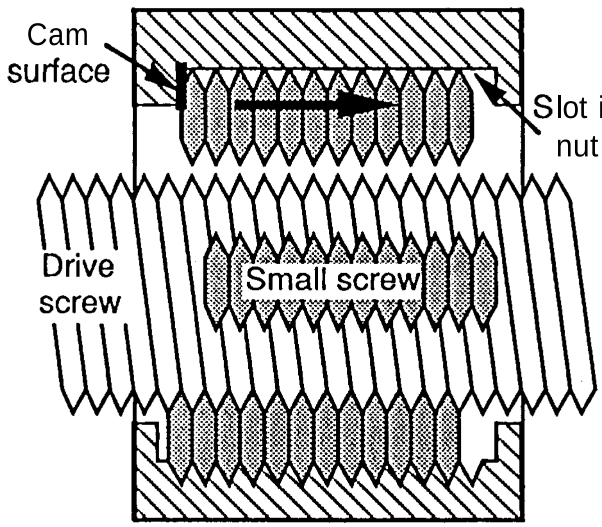

Figure 10b. A roller screw

A more precise device for converting rotary to linear motion is the roller screw. Rather than using balls between the screw and nut to reduce friction, the roller screw uses small screws as shown in figure 1. Ob.

This arrangement allows finer pitches (as fine as 1 mm/rev) while being able to support higher loads due to the large contact area between the recirculating screws and the nut or drive screw. The small screws recirculate as the nut rotates. There is a slot inside the nut which the small screws enter, and a cam surface in the nut pushes the screw back one thread, at which potnt it meshes with the main screw again. Roller screws normally exhibit some backlash, but pre-loaded units without backlash are available. Roller screws are ideal for fine position control, high load applications; they are used in the Keck telescope to articulate primary mirrors. Like ball screws, the recirculation in the roller screw generates torque disturbances that make it less desirable for rate control applications.

2.3.5. Transmission comparison.

The table below summarizes the features of the transmission elements discussed:

	spur Gears	**Harmonic Drive**	**Ball/roller screw**	**Band drive**
Mechanical Advantage	≤ 6:1 per pass, up to ~100:1 for planetary gear train	60:1 to 200:1	Up to 2 mm/rev for ball screw, up to 1 mm/rev for roller screw	Approximately the same as for spur gears
Lost motion	Anti-backlash gears available	Gear error	Thread error, Can be preloaded to eliminate backlash	Virtually none
Friction	Depends on ratio, no. of passes.	-0.05 Nm	Depends on preload	Extremely low
Torque disturbances	Significant	Highest	Significant	Virtually none
Life	Decreases with mechanical advantage	Slightly less than that of ball bearing	Comparable to that of ball bearings	Limited by bearings

Table 3. Transmission comparison

2.4 Sensors

A wide variety of displacement sensors are used in space mechanisms. We will compare the resolver. Inductosyn, LVDT, optical encoder, and potentiometer.

2.4.1. Resolver/Inductosyn

The resolver and Inductosyn, shown in figure 11, work on the same principle. The resolver looks very much like a motor, consisting of a rotating excitation coil which is inductively coupled to two fixed coils, 90° out of phase with each other. The amplitude of the signals in the pick-up coils are proportional to the sine and cosine of the excitation coil angle. Circuitry demodulates the pickup signal and converts the sine and cosine into a digital angle. The excitation signal can be brought to the rotating coil over wires which cross the rotating interface, or by a rotary transformer (at the expense of more power) if continuous rotation is required. The conversion from sine and cosine to angle is accomplished with a hybrid chip called a resolver-to-digital converter, with resolution as fine as one part in 216. However, the accuracy is limited by errors in the iterative conversion process, as well as errors in the resolver itself. In order to improve accuracy, multi-pole (also called "multi-speed") resolvers are used which generate several sinusoids per revolution. Each sine can be accurately digitized to typically one part in 212, but if there are sixteen sinusoids per revolution, the accuracy is now one part in 216. In practice, the output of a multi-pole resolver is correlated to that of a single pole resolver so that counting sinusoids is not required for absolute angular knowledge.

The Inductosyn, built by Farrand Controls of Valhalla, NY, exploits printed circuit technology to expand on the multi-pole principle. It consists of a pair of disks with windings printed on their faces. This design allows as many as 1024 poles to be put on an Inductosyn, enabling much higher accuracy and resolution than is possible with a wound resolver. The printed disks are also more compact and lighter weight than an equivalent resolver. However, the printed windings do have a much weaker Inductive coupling than a wound resolver, therefore Inductosyns require more power and tighter mechanical alignment. Linear motion Inductosyns are also available.

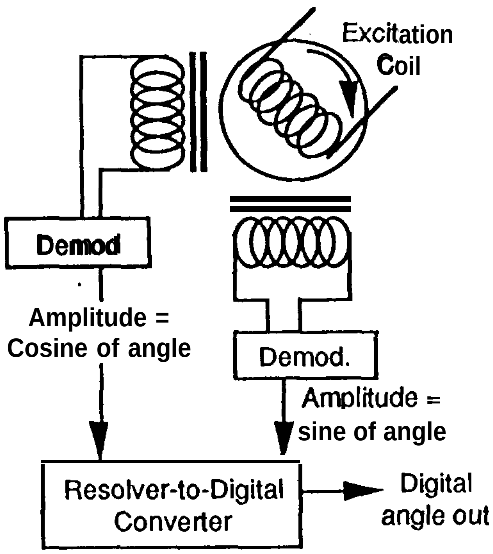

Figure 11. Schematic diagram of a single-pole resolver

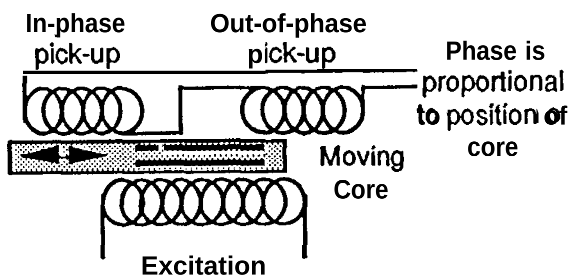

Figure 12. Schematic diagram of an LVDT.

2.4.2. LVDT

The Linear-Variable Differential Transformer (LVDT) is similar to the resolver, in that it consists of an excitation coil inductively coupled to a patr of pick-up coils, out of phase with each other, as shown in figure 12. In an LVDT, both the excitation and pick-up coils are fixed, and an iron core moves. The pick-up coils are connected in series. When the core is at null posit-ton, the induced signals cancel and the output is zero. At either side of null, the amplitude of the induced signal is proportional to position. The phase of the tnduced signal indicates which side of null the core is on. The LVDT is used to measure linear motion, but its cousin the RVDT measures rotary motton over limited angles.

2.4.3. Optical encoder

The optical encoder is another common displacement sensing device. A light source, usually an LED, is focused through a patterned disk onto a photodetector. As the disk rotates, it modulates the light that falls on the detector; each pulse indicates an increment of motion. An incremental encoder will typically have another track with one pulse per revolution as an absolute reference. Rotation direction is determined by sensing pulses with two sensors, 1/2 pulse out of phase. This is called quadrature, and it also increases the encoder's resolution by a factor of four. Absolute knowledge can be determined by using a series of binary-encoded tracks, one for each bit in the binary angle word, Encoders are available with resolution up to one part in 218, but even higher resolution and accuracy can be achieved by interpolating between pulses with a technique similar to that used by a resolver. The encoder disk must be precisely aligned relative to its optics, therefore encoders typically require their own bearings, and are connected to a motor through a flexible coupling so as not to overconstrain the motor bearings. The flexible coupling introduces some angular error. Encoder life is limited by the life of its light source and by the life of the photodetectors. Linear motion optical encoders are also available. BEI Motion Systems Company is a major source of flight-qualified encoders.

2.4.4. Potentiometer

The potentiometer is perhaps the simplest displacement sensor. It consists of a brush which rides on a resistive track, the resistance varying with angle. For space applications, potentiometers are somewhat of a reliability risk, because of the fragility of the brushes under launch loads; and because of the limited life due to wear. Potentiometers have limited accuracy, and their signal is frequently noisy, especially as wear debris accumulates. They are also a source of friction. However, when accuracy requirements are sufficiently lax, such as for solar panel pointing, the potentiometer can be an inexpensive sensor option.

2.4.5. Sensor Comparison.

Features of the sensors are summarized in the table below:

	Resolver	Inductosyn	Encoder	Potentiometer
Accuracy	<100 urad	<1 urad	25 urad	10 mrad
Mass	Highest	Low	High	Lowest
Power	High	Highest	Low	Lowest
Integration with motor	Simplest	Requires tighter alignment than resolver	Separate assembly connected by flexible coupling	Separate assembly connected by flexible coupling
Reliability	High	High	Limited by LED	Subject to electrical noise and wear
Signal transfer	Requires rotary transformer or leads	Requires rotary transformer or leads	None	Requires brushes
output	Analog sine & cosine or digital word	Digital word	Digital word or quadrature pulses	Analog
Electronics complexity	Complex	Most complex	Simple	Simplest

Table 4. Sensor comparison

2.5. Signal transfer

Getting signals and power from one side of a moving interface to the other is an often neglected detail that is critical to the reliability and performance of an actuator. Several options will be discussed here.

2.5.1. Cable bundle

The simplest option is to simply leave slack in a cable bundle that crosses a joint. However, cable bundles have unpredictable, non-linear stiffness properties. Typically, stiffness increases with decreasing temperature, and the torque versus angle profile will include a large hysteresis loop similar to that of Dahl friction. All of these properties complicate the task of sizing an actuator to articulate the joint under all conditions. On the Voyager scan platform, tie-wraps had to be cut away from the cable bundle at the last minute to reduce its stiffness to the point where the actuator could move the platform.

On the Hubble telescope, slack in one of the antenna cables was insufficient to allow full articulation of the antenna.

2.5.2. Flex-tape assembly

A better way to control the stiffness of the signal transfer is to use a ribbon cable or "flex-tape" designed for flexibility. One type consists of copper foil conductors inside a Kapton film sandwich. Multi-layer tapes are available which can provide signal shielding. The tape can be wound like a clock-spring and installed in a housing integral with the actuator, so that actuator properlies, including the load due to signal transfer, can be tested in one assembly. Stiffness can be further reduced by using a pair of tapes, one wound clockwise and the other wound counterclockwise. This type of assembly still exhibits non-linear stiffness and torque hysteresis, but with amplitudes that are much more benign. The range of motion for this assembly can exceed 360°. One supplier of flex-tape assemblies is Electro-Tec of Blacksburg, VA.

2.5.3. Slip rings and roll rings

Obviously, cables can not be used across continuously-rotating joints. Here, the most common approach is to use slip rings. Slip rings consist of a set of brushes which contact a set of tracks on a rotating disk or drum. Brush and track materials are typically a silver-dry lubricant composite, or wet-lubricated gold-on-gold. Slip ring assemblies exhibit coulomb friction, and they can be a significant reliability risk. Their lives are limited by wear, and floating wear debris can cause short circuits across adjacent rings (which led to the demise of the SeaSat mission). For that reason, high and low signals are placed at opposite sides of the assembly, and slip rings must be run-in for several thousand revolutions prior to use to reduce the rate of debris generation, Slip rings can also generate electrical noise. so they are not recommended for digital signal transfer. Slip ring assemblies are made by Electro-Tec and Litton Poly-Scientific, both of Blacksburg, VA.

Honeywell Satellite Systems in Glendale, AZ has developed an assembly that uses rolling rings, rather than rubbing brushes, to transmit signals. Obviously, wear and friction is greatly reduced. A roll-ring assembly is scheduled for use on the Space Station D.

2.5.4. Rotary transformers

Ideally, we would like to transmit signals without mechanical contact, and that is what a rotary transformer does. This is a device customized for the particular frequency of the signal transmitted, and it can only be used for digital or other high-frequency signals. AC power can be transmitted this way, but with poor efficiency. In the Galileo spacecraft, slip rings were used for power and low-frequency analog signal transfer, and rotary transformed were used for digital signal transfer.

2.5.6. Signal Transfer Comparison.

Features of the four signal transfer techniques are summarized in the table below:

	Cable	Flex-tape	Slip rings	Rotary transformer
Range of motion	<180°	<360°	Continuous	Continuous
Mechanical impedance	Non-linear stiffness, hysteresis	Low non-linear stiffness, hysteresis	Coulomb friction	No mechanical contact
Life	Limited by fatigue	Limited by fatigue, $>10^7$ cycles	Limited by wear. $>10^7$ Cycles	Unlimited
Signal compatibility	Unlimited	Unlimited	Best for low-bandwidth analog signals	Inefficient for power transfer. Limited to narrow frequency range
Reliability	Stiffness difficult to predict, can hang	High	Wear debris can cause shorts	High

Table 5. Signal transfer comparison

3. DESIGN EXAMPLES

The following are some examples of actuator applications applicable to interferometers, and the design solution that was used or proposed:

3.1 Inertial pointing, position stability at rate.

Examples of this problem are pointing the scan platform on the Galileo spacecraft, and on the Cassini spacecraft (before the articulated platform was deleted as a cost-saving measure). On both of these spacecraft, the inertial rate sensor is located on the platform, and the platform must be pointed at an inertially-stable target (a planet or moon) regardless of the spacecraft motion. The intent is to provide better platform pointing with the actuator than can be obtained by pointing the spacecraft with reaction wheels and/or thrusters.

The solution is to do everything possible to minimize the mechanical coupling across the rotating joint. This means using a direct-drive actuator driven by a DC brushless motor. The motor is sinusoidally-commutated to minimize torque ripple. Any non-contacting angle sensor can be selected, based on accuracy requirements. Since the inertial angle is determined by the gyro on the platform, the angle sensor accuracy is based on how well the spacecraft angle, relative to the gyro, must be known (i.e. for antenna pointing). The Galileo scan actuator used a 16-bit absolute optical encoder, while we proposed a multi-pole resolver for Cassini (for longer life). Both actuators used redundant motors and redundant sensors (two read-heads on a single disk for the encoder), and both were supported on a single pair of angular contact bearings. Signal transfer for Galileo was accomplished with a flex-tape assembly built into the actuator housing. For Cassini, two versions of the actuator were required; one for limited-angle motion of the scan platform and one for continuous rotation of another science platform. To minimize cost, we proposed the modular approach shown in figure 13. The same drive module (motor, resolver, bearings) could be configured with a flex-tape module or slip ring-rotary transformer module. The drive module had to be somewhat overdesigned to support the loads and friction of either application, but, on paper at least, it promised millions of dollars in savings over designing two separate actuators.

The concept of minimizing mechanical contact can be extended for more demanding applications. One technique that has been used successfully is to nest a set of flexures in series with ball bearings, so that fine motion can be accomplished without having to compensate for ball bearing friction. The flexure can be driven with its own vernier single-pole torquer. Alternately, magnetic bearings could replace the mechanical bearings. Of course, there is little point to reducing bearing friction if the most significant mechanical coupling is due to signal transfer; the actuator designer must work with other subsystems to minimize the number of signals if that affects actuator performance.

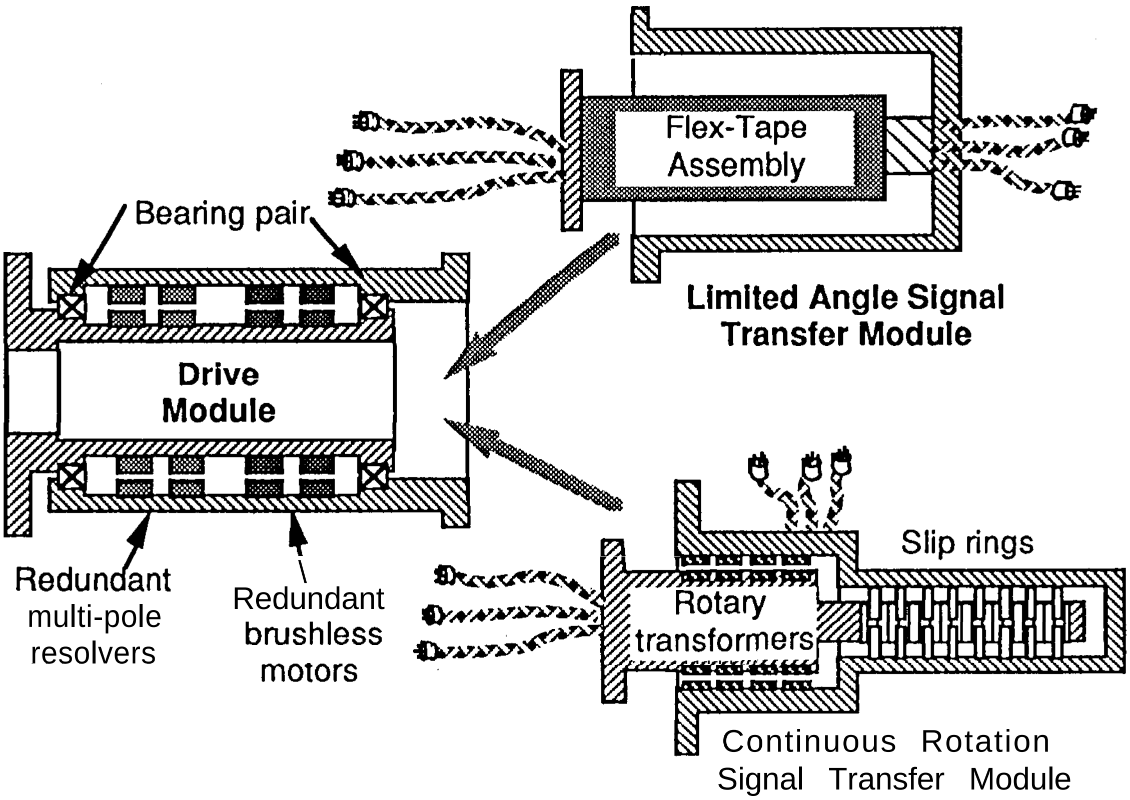

Figure 13. Modular direct drive actuator proposed for Cassini.

<u>3.2 Body-relative pointing, low to moderate precision.</u>

This type of application assumes that the spacecraft is pointed accurately enough to be used as a reference for the actuator. Examples include solar panel and antenna actuators. For applications like these where rate stability is not important (except perhaps in limiting torque disturbances that could affect spacecraft pointing), a stepper motor can be used to save electronics complexity and cost, and a transmission can be used to reduce motor mass and save power (mechanical power losses increase, but the motor can operate at a more efficient speed, resulting in a net savings). Several vendors make an actuator like this consisting of a stepper motor and a harmonic drive, It is frequently configured with a resolver or potentiometer (potentiometers were used on the Magellan and TOPEX solar array drives). We have proposed this type of actuator for the POINTS solar array drive, configured as shown in figure 14. The actuator is configured as a separable assembly from the joint bearings and flex-tape assembly, so that the actuator can be tested independently of the spacecraft structure. The actuator can also be specified and procured without a complete knowledge of launch loads on the joint. This reduces procurement costs by increasing the likelihood that a pre-existing actuator design will be acceptable. Manufacturers of this type of actuator include Ball Aerospace, Honeywell Satellite Systems, Schaeffer Magnetics, and TRW.

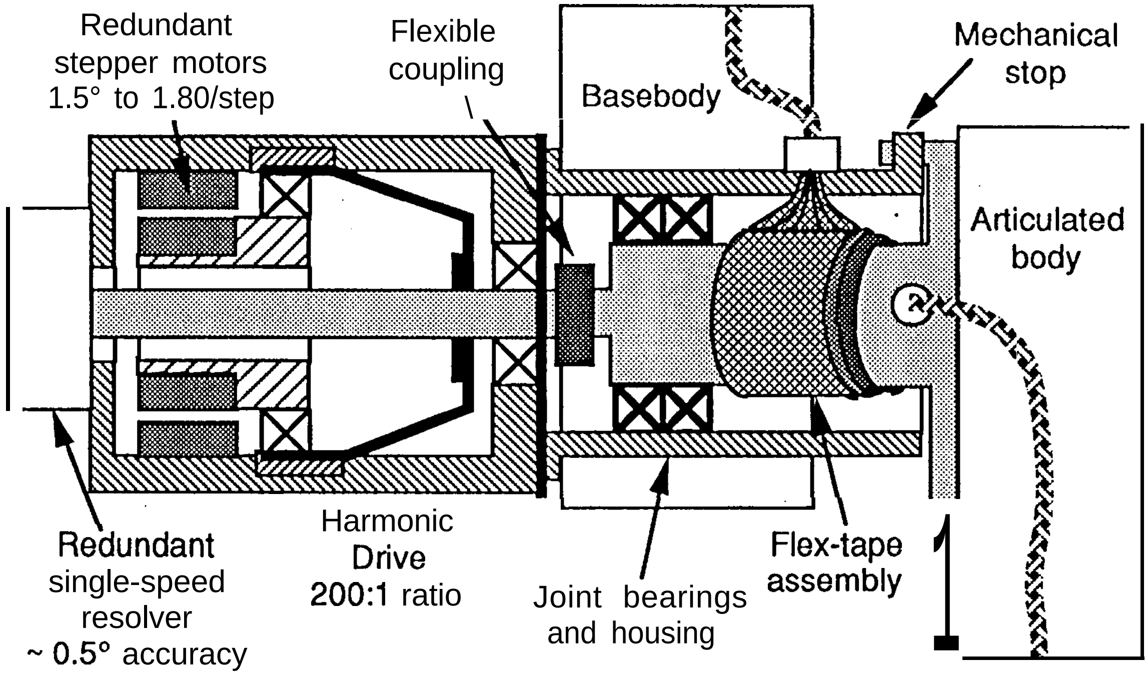

Figure 14. Actuator proposed for POINTS solar array articulation.

3.3 Body-relative pointing, high-precision.

This example applies specifically to the POINTS articulated interferometer. The POINTS instrument consists of two large optical benches (approximately 2 m square) which are pointed relative to each other in one axis. The range of motion is limited to ±50 rnrads, but the pointing accuracy requirement is ±2.4 prads. Rate stability is not a requirement, simplifying the task. The limited range of motion allows the use of flexures to constrain the articulation axis. Flexures are preferred for their predictable reliability.

A significant mechanical advantage will be required to reduce large, easily-controllable motor motion to the Wad range, and the challenge is to make the transmission as simple as possible. Figure 15 shows the proposed actuator solution, which is a linkage driven by a linear actuator. For starters, we take advantage of the size of the optical bench and use the largest possible moment arm. We propose to locate the linear actuator 1 m from the articulation axis. To generate ±50 mrads of optical bench motion, the actuator's range of motion must be ±5 cm. The linear actuator consists of a brushless motor driving a roller screw with a 1 mm/rev pitch. Torque disturbances due to roller screw recirculation are not a concern because we have no rate stability requirement. To control optical bench motion to 2.4 µrads, the motor rotation must be controlled to 15 mrads, a relatively simple task. The motor can be square-wave commutated, again because we are unconcerned with rate stability. If we had a larger moment arm or the accuracy requirement were reduced, we could use a stepper motor and further simplify the system, Flex tapes will be used to transfer signals and power from one optical bench to the other.

None of the sensors discussed in this paper can provide feedback accurate to 2.4 Wads, so the instrument's laser interferometer metrology is used instead. However, we proposed that an LVDT be built into the actuator for low-accuracy control during testing and safing, when the laser metrology might not be available.

The linear actuator must be attached to the optical benches such that it does not overconstrain the flexures defining the axis of rotation; we propose the bearing arrangement shown in figure WW. An equivalent flexure system could also be used. A similar linear actuator, using a brush motor and a coarser ball screw, was used on the Viking Orbiters to control engine pointing, and the design will be used again for Cassini spacecraft engine pointing.

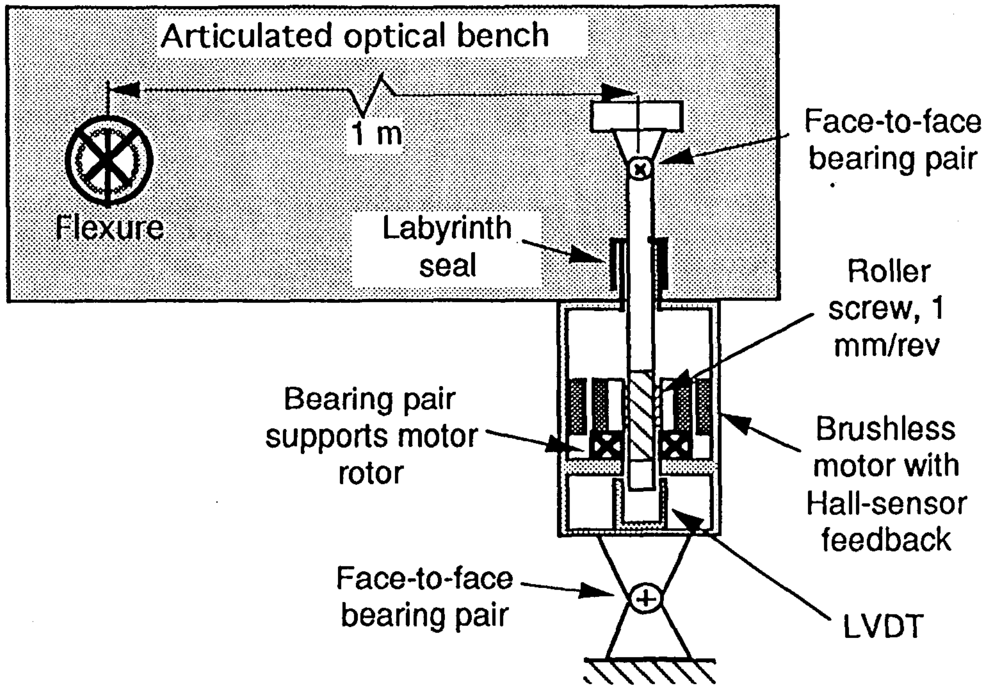

Figure 15. Linear actuator for POINTS

4. HOW TO SPECIFY AN ACTUATOR

Now that you are armed with a complete technical understanding of how actuators work, you are ready to communicate your desires to the actuator engineer. Here are some tips for specifying actuators

4.1 Rules of Thumb

Get an actuator engineer involved in the system design as early as possible. This will insure that you have realistic expectations of the system performance, reliability, and cost while you still have flexibility in defining the system.

To minimize cost, try to accommodate existing actuator designs as much as possible, Even small changes from an existing design can double the price of a device. Unfortunately, most actuator applications are so specialized that existing devices are incompatible with requirements. In that case, try to modularize the design such that significant parts of it can come from existing designs. An example of this approach is the stepper-harmonic drive actuator described above (an existing design) coupled with a separate custom output bearing and flex-tape assembly. On the other hand, don't specify a device with heritage unless you thoroughly understand its capabilities. In the end, it could cost more to redesign the spacecraft around an inexpensive but inappropriate actuator.

Use the minimum number of requirements possible to define a function. Additional constraints cost money and inhibit the designer. Don't mix "what to do" with "how to do it."

Use components with predictable behavior; tests and analyses to prove compliance with requirements are major cost drivers.

Keep it simple; complexity equals cost.

4.2 Requirement Tips

Define position and rate performance terms precisely, and preferably with graphics. A requirement such as "rate stability: 0.5 mrads" is meaningless. Is it plus-and-minus or peak-to-peak? Is it RMS, or 3-sigma? Does the frequency of the rate disturbance matter? Interpretation of a requirement should not be left as an exercise to the reader.

Don't forget that an actuator inflicts loads not only on the body it is pointing, "but on the rest of the spacecraft. If necessary, specify an allowable disturbance spectrum. Generating such a requirement will typically require a dynamic structural model of the entire spacecraft.

Launch loads, not operating loads, size most actuators. Being conservative in determining launch-load requirements can save analysis costs, but may result in more mass and worse pointing performance than is necessary.

Does the actuator have to constrain platform motion in the controlled degree of freedom during launch, or is there a latching mechanism? Actuator torque capability or internal friction may have to be selected for launch instead of for operation. A latching mechanism can reduce the load on the actuator, improving performance, but at the expense of another mechanism with mass, cost, and failure modes.

5. SUMMARY

Spacecraft configuration, performance, and cost are heavily dependent on actuator selection. Having the knowledge to make good actuator-related trades early in a program can avoid the need for radical, painful design changes down the road. The Cassini project is a good case study of this process. After spending years developing a spacecraft with two articulated platforms, new cost constraints dictated a new design without articulation. The net savings was on the order of tens of millions of dollars, but at the expense of reduced science quality. Knowing the cost-benefit trade-off for articulation, if it had been possible to anticipate future budget cuts, even more money could have been saved by avoiding years of work on the initial "wrong" design. Maybe enough to pay for one articulated platform? The point is that moving parts on a spacecraft are complicated and expensive, and their benefits must be carefully appraised. Consult an actuator engineer early and often to evaluate your spacecraft design options.

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